THE EFFECT OF A 180^{o} BEND ON TURBULENT HEAT TRANSFER COEFFICIENT IN A PIPE

By

MAHMOOD MOSHFEGHIAN

Bachelor of Science

Oklahoma State University

Stillwater, Oklahoma

1974

Submitted to the Faculty of the Graduate College
of the Oklahoma State University
in partial fulfillment of the requirements
for the Degree of
MASTER OF SCIENCE
July, 1975

THE EFFECT OF A 180° BEND ON TURBULENT

HEAT TRANSFER COEFFICIENT

IN A PIPE

Thesis Approved:

Kenneth J. Bell
Thesis Adviser

Jerald D. Parker

John H. Erbar

D. D. Durham
Dean of the Graduate College

PREFACE

Heat transfer when water flow through a 180^{o} bend tube was studied. A Reynolds range from 7,300 to 27,000 was investigated for a bend radius of 9.875 inches and 7/8-inch OD x 0.052-inch wall thickness. The test section was Inconel tube and was heated electrically by passing DC current through the tube wall.

I am very grateful to my adviser, Dr. K. J. Bell for his expert guidance and much valued counsel during the course of my study. I am also grateful to my Advisory Committee which consisted of Drs. J. H. Erbar and J. D. Parker, for their very helpful criticisms and suggestions given. I would like also to extend my appreciation to Dr. M. N. Farukki for providing me the computer program.

I thank the School of Chemical Engineering of Oklahoma State University for providing me financial assistance during the courese of my study.

I am grateful to Mr. E. E. McCroskey, the storeroom manager for his assistance in the fabrication of the equipment. I also thank Mr. Reza Ahmadnia and Mr. Ahmad Moshfeghian for helping me to draw the computer flow charts and diagrams of this work.

Finally, I would like to express my appreciation to my father, Mr. Ali A. Moshfeghian for encouraging and providing me financial support through my education.

TABLE OF CONTENTS

Chapter Page

I. INTRODUCTION. 1

II. LITERATURE SURVEY 3

III. EQUIPMENT 6

Loop Description 6
Test Section Description 6
Temperature Measurements 8
Pressure Measurements. 11
Flow Measurements. 11
Electrical Measurements. 11

IV. EXPERIMENTAL PROCEDURE. 12

Thermocouple Calibration 12
Manometer Calibration. 13
Rotameter Calibration. 13
Heat Loss Calibration. 13
Loop Operating Procedure 13

V. EXPERIMENTAL RESULTS AND DISCUSSION 15

Local Results. 15
Average Results. 26
Error Analysis 32

VII. CONCLUSIONS AND RECOMMENDATION. 33

BIBLIOGRAPHY. 35

APPENDIX A. 36

APPENDIX B. 41

APPENDIX C. 52

APPENDIX D. 55

Chapter Page

APPENDIX E. 67

APPENDIX F. 78

LIST OF TABLES

Table		Page
I.	Ratio of Heat Transfer Coefficients from Present Work to Those Predicted by Literature.	28
II.	Calibration Data for Rotameter.	37
III.	Calibration Data for Inlet and Outlet Thermocouples During In-Situ Calibration of Surface Thermocouples on the Test Section	38
IV.	Calibration Data for Heat Loss from Test Section.	39
V.	Calibration Data for Calibration of Outside Surface Thermocouples	40
VI.	Run 209 - Outside Surface Temperature ^{o}F.	59
VII.	Run 209 - Calculated Inside Surface Temperatures, ^{o}F. . .	60
VIII.	Run 209 - Radial Heat Flux for Inside Surface, $Btu/Hr\text{-}Ft^2$.	61
XI.	Peripheral Heat Transfer Coefficients at Station 1. . . .	65

LIST OF FIGURES

Figure		Page
1.	Secondary Flow in Curved Tube	2
2.	Test Section Set-Up and Flow Direction Used by Lis (1), Ede (2), and Tailby (3)	4
3.	Heat Transfer Loop (5)	7
4.	Inconel Test Section (5)	9
5.	Peripheral Distribution of Heat Transfer Coefficients, Run 207	17
6.	Peripheral Distribution of Heat Transfer Coefficients, Run 207	18
7.	Peripheral Distribution of Heat Transfer Coefficients, Run 207	19
8.	Peripheral Distribution of Heat Transfer Coefficients, Run 210	20
9.	Peripheral Distribution of Heat Transfer Coefficients, Run 210	21
10.	Peripheral Distribution of Heat Transfer Coefficients, Run 210	22
11.	Peripheral Distribution of Heat Transfer Coefficients, Run 215	23
12.	Peripheral Distribution of Heat Transfer Coefficients, Run 215	24
13.	Peripheral Distribution of Heat Transfer Coefficients, Run 215	25
14.	$\overline{Nu}$ Number as a Function of Re and Position in Test Section	31

NOMENCLATURE

A	- area
C_p	- specific heat, Btu/(lbm-^{o}F)
d_i	- inside tube diameter, inches
d_o	- outside tube diameter, inches
G	- mass velocity, lbm/(hr-ft^2)
h	- peripheral heat transfer coefficient, Btu/(hr-ft^2-^{o}F)
$\bar{h}$	- averaged heat transfer coefficient around tube, Btu/(hr-ft^2-^{o}F)
j	- $= NuPr^{-0.4}(\mu_w/\mu_i)^{-0.14}$, dimensionless number
K	- thermal conductivity, Btu/(hr-ft-^{o}F)
Nu	- Nusselt number, hd_i/K dimensionless number
Pr	- Prandtl number, $C_p\mu/K$, dimensionless number
Q, q	- heat flow rate, Btu/hr
Q/A, q/A	- heat flux, Btu/(hr-ft^2)
R	- bend radius, measured to the longitudinal axis of tube, inches
r	- tube radius, inches
T	- fluid temperature, ^{o}F or ^{o}C

t_b - bulk fluid temperature, ^oF

t_{w_i} - inside wall temperature, ^oF

v - vluid velocity, ft/sec.

w - mass flow rate of fluid, lbm/hr

x - axial distance along the tube from the entrance of straight section downstream of the bend, inches

X - axial distance along the tube from the mid-point of 180^o bend, inches

Greek Letters

μ - fluid viscosity at fluid bulk temperature, centipoise or lbm;(f5-hr)

μ_w - fluid viscosity at wall temperature, centipoise, or lbm/(ft-hr)

ρ - electrical resistivity, (ohms-in^2)/in., or density lbm/ft^3 or gm/ml

Subscripts

avg - average

b - bulk fluid

i - inside of tube

in - inlet of test section

o - outside of tube

out - outlet of test section

w - wall

CHAPTER I

INTRODUCTION

U-tubes (two lengths of straight tube joined by a 180° bend at one end) are commonly used in a large variety of tubular heat exchangers: for example, a continuous length of tubing may be bent back and forth across a duct. U-tubes are also used in shell-and-tube heat exchangers having entry and exit headers at the same end.

For a fully developed flow inside a straight horizontal tube, it is expected that the temperature profile would be symmetric about a horizontal axis passing through the center of the tube. However, for a fluid flowing axially in a curved tube, it is known that an induced secondary flow, as shown in Figure 1, is superimposed on the primary flow. Therefore, one would expect that the secondary flow enhances the heat transfer in the direction normal to the tube axis. Additionally, one would expect the heat transfer coefficients around the tube would be far from uniform, being probably highest on the outer side (position 5 in Figure 1) and lowest on the inner side (position 1). These results have been generally verified by experiments reported in the literature.

The present investigation was undertaken to study the effect of a 180° bend on heat transfer to water in a tube.

Experiments were made with water flowing through an electrically-heated tube to explore the variation of the local tube-to-water heat transfer coefficient upstream of the 180° bend, in the bend, and downstream from the bend. The test section used was an Inconel tube of 7/8-inch OD x 0.052-inch wall thickness and a bend radius to the axis of 9.875 inches.

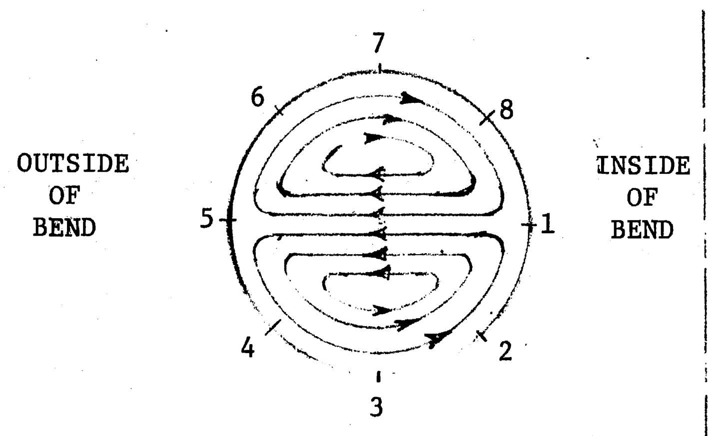


Figure 1. Secondary Flow in Curved Tube

CHAPTER II

LITERATURE SURVEY

Despite the obvious importance of the effect of 180° bend, there is only a limited number of investigations reported in the literature.

Lis et al. (1) investigated the turbulent heat transfer in a vertical pipe, with upwards flow, preceded by a 180° bend (Figure 2). Based on their experimental data, they state the general shape of the curve of heat transfer coefficient against non-dimensional axial position (x/d) did not change very much with Reynolds number, and a simplified correlation for averaged heat transfer coefficient around the tube is as follows:

$$j_x = 0.0239\ Re_x\ (x/d)^{-0.064}\ (R/r)^{-0.062} \tag{1}$$

where

$1 \leq x/d \leq 15$, $8000 \leq Re_x \leq 94000$, $5.5 \leq Pr \leq 9.7$, and $2/1 \leq R/r \leq 4/1$.

Ede (2) studied the effect of a 180° bend on heat transfer to water in a vertical tube. He reported that the flow visualization experiments for the laminar region have shown that circulation continues vigorously all around the bend, but appears to stop almost immediately once the straight part has been reached. For the turbulent region, he concluded that the velocity of the liquid near the outside of the bend is much higher than that near the inside, and

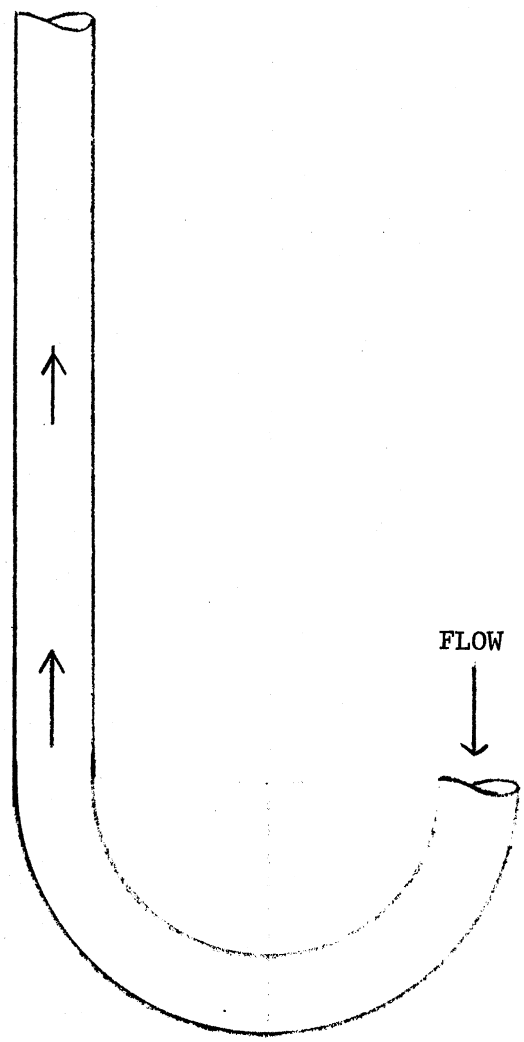


Figure 2. Test Section Set-Up and Flow Direction Used by Lis (1), Ede (2), and Tailby (3)

a secondary circulation develops; these conditions may extend for a considerable distance downstream. These secondary effects give rise to higher heat transfer coefficients than normal on the outside of the bend and lower on the inside, the average effect being an increase. For Reynolds numbers in the neighborhood of 5000 to 10,000, the possibility that an incipient laminar flow may develop has been tentatively advanced as an explanation of the production of unusually low heat-transfer coefficients.

Tailby et al. (3) also investigated the influence of 90° and 180° pipe bends on heat transfer from an internally flowing air stream in a vertical pipe with upwards flow. For a 180° bend, they also have derived an empirical correlation which includes the effect of curvature on heat transfer coefficient

$$Nu = 0.0341\ Re^{0.82} Pr^{0.4} (x/d)^{-0.04} (R/r)^{-0.11} \tag{2}$$

where

$$7 \leq x/d \leq 30\ ,\ 4 \leq R/r \leq 14\ ,\ \text{and}\ 10{,}000 \leq Re \leq 50{,}000$$

Finally, for air, the turbulent heat transfer rate and wall temperature distribution in rectangular ducts (aspect ratio 10) with a 180° bend were studied experimentally by Yang (4). He also concluded that the heat transfer coefficient is higher at the outside wall compared to that of the inside wall of the bend.

CHAPTER III

EQUIPMENT

The equipment was assembled by M. N. Farukhi, and most parts of this chapter and the following chapter are taken from his Ph.D. thesis (5).

Loop Description

The loop used here was essentially the same as that employed by Farukhi (5); there were some minor changes. A schematic diagram of the loop is shown in Figure 3. Liquid water was supplied to the system from a Colora® constant temperature bath which included a temperature controller, a mixer, and a small centrifugal pump. The water injection system was made from 1/2 inch copper, brass, and stainless steel tubing and fittings. Water from the bath was pumped by a 3 HP centrifugal pump through a 1/2 inch Manatrol® needle valve, and a Fischer and Porter® rotameter to the test section. Part of the water supplied by the pump was returned to the bath via a bypass line. The bypass flow rate was controlled by a 1/4 inch Manatrol® needle valve. From the test section the hot water was sent to a small heat exchanger to cool down the water to bath temperature.

Test Section Description

The test section was made from 7/8 inch OD x 0.052 inch wall Inconel 600 seamless tube. Inconel 600 was used because of its high

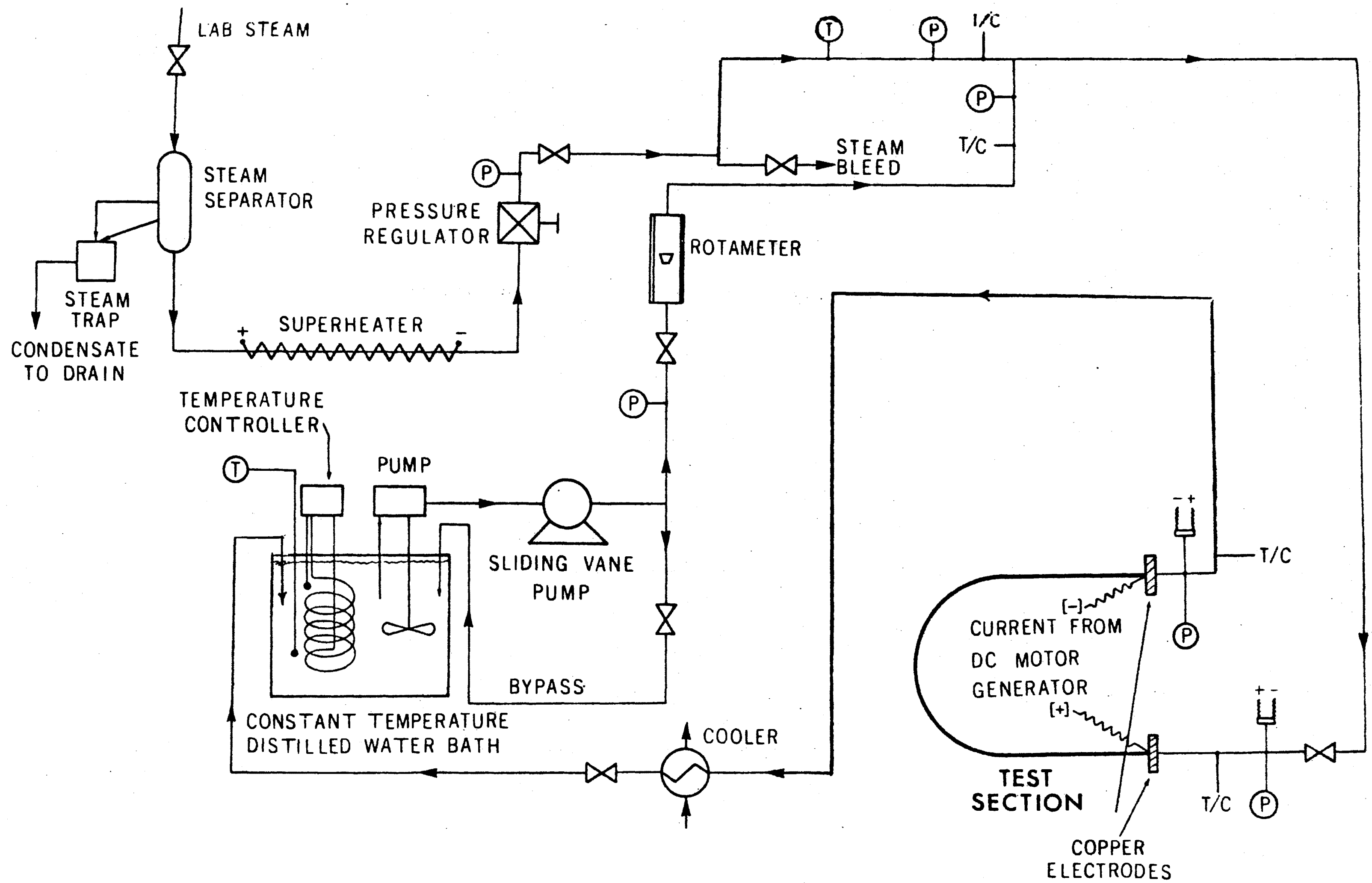


Figure 3. Heat Transfer Loop (5)

electrical resistance. The test section has a bend radius of 9.875 inches measured to the longitudinal axis of the tube. The test section is shown in Figure 4. The test section was annealed after bending so that stress relieving would be avoided during testing.

The test section was thermally insulated by wrapping it with several layers of bonded fiberglass tape and two inches of fiberglass sheets. The outside surface of insulation was then wrapped with aluminum foil so that radiation losses would be minimized.

The test section was also electrically isolated from the loop by connecting it with a short piece of neoprene tubing at each end.

Four pressure taps with 1/16 inch hole were silver-soldered to the test section. The taps were electrically isolated from the recording instrument by connecting them with silicone rubber tubing.

The test section was heated by passing DC current generated by a Linconlnweld® SA-750 motor generator set. Two thick copper bars were silver-soldered to the test section to serve as electrodes for connection to the motor generator.

Temperature Measurements

The test section was fitted with 88 thermocouples; their locations are shown in Figure 4. All the thermocouples were made from 30 AWG wire since the thinner wire was more flexible and had smaller induced stress and conduction losses.

The thermocouples were glued and mounted to the test section by the procedure outlined below.

1. A bead of Saureisen® No. 33 cement was placed at each thermocouple location.

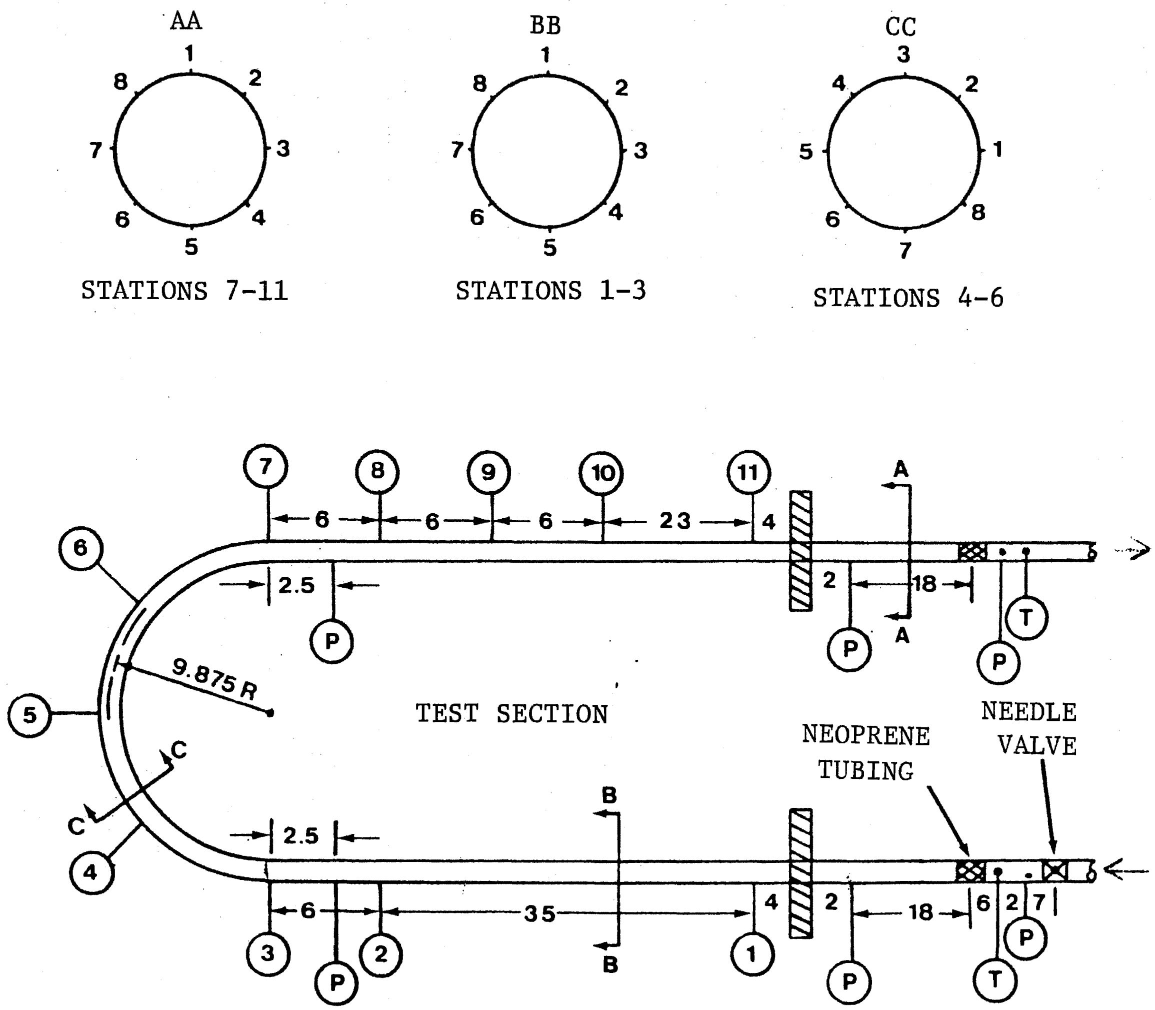


Figure 4. Inconel Test Section (5)

2. After the bead dried completely it was sanded flat to about 1/2 mm thickness.

3. The hot junction of thermocouple wire was placed at the designated location and the wire was temporarily held in place with adhesive tape.

4. Another bead of cement was then placed on the hot junction so that the thermocouple was completely glued to the surface.

5. After the cement dried the thermocouple was held to the tube by a strap of wire. A three inch span was allowed between the hot junction and the strap of wire holding the thermocouple to the tube to minimize conduction errors.

The 88 thermocouples were placed at eleven stations along the test section. At each station, 8 thermocouples were located peripherally around the tube. The positions of stations and their thermocouple locations for upstream, in the bend, and downstream are shown in Figure 4.

The following fluid temperatures were also measured for each experimental run (see Figure 4 for their locations in the loop).

1. Water inlet temperature.

2. Water outlet temperature.

The inlet and outlet water temperatures were measured by ungrounded, sheathed thermocouples with hot junctions exposed to the fluid flow. The thermocouple assemblies were attached to the inlet and outlet lines by Swagelok® fittings.

All the surface thermocouples were connected to an array of barrier strips which in turn were connected to 11 rotary switches. The terminal lugs used on the barrier strips were made of either iron or constantan and this feature eliminated any problems associated with creation of new thermocouples due to variation in room temperature. The rotary switches, however, were mounted on a panel and the connections were enclosed in a constant temperature box. The outputs from rotary switches were brought to a master rotary switch which was connected to a Leeds and Northrup Numatron®. The Numatron incorporated a reference junction compensator and the output was displayed in digital form in degrees Fahrenheit. The Numatron was calibrated prior to usage according to manufacturer's specification.

Pressure Measurements

The six pressure taps, as shown in Figure 4, were connected to a manifold by a series of Whitey valves. The switching system was connected in such a manner that any of the six taps could be activated and read on a Meriam® U-type manometer against atmospheric pressure.

Flow Measurements

The flow rate of water was measured by reading the output on a calibrated Fischer and Porter® rotameter.

Electrical Measurements

The current to the test section was measured by a Weston® Model 931 ammeter in conjunction with a 50 MW shunt. The voltage drop across the test section was measured by a Weston® Model 931 voltmeter.

CHAPTER IV

EXPERIMENTAL PROCEDURE

Thermocouple Calibration

All surface thermocouples were calibrated in situ by using saturated steam as reference temperature. To calibrate the thermocouples, the laboratory steam was allowed to go through a steam separator and liquid water was trapped out of stream. Then the saturated dry steam at atmospheric pressure was sent to the test section and the thermocouple outputs were recorded after a run time of approximately eight hours. The calibration runs were performed two times. During the runs, the atmospheric pressure and room temperature were also recorded so that the surface temperatures could be compared with saturated steam temperature. During calibration, the pressure in test section at six different locations was measured by a manometer against atmospheric pressure. The inlet and outlet water thermocouples were also calibrated during the calibration runs of surface thermocouples by comparing the inlet and outlet thermocouples reading with steam temperature at atmospheric pressure.

The thermocouple readings for all runs were corrected by assuming that the conduction losses were proportional to the difference between the thermocouple reading and room temperature for each run as compared to the difference during calibration (see Appendix A for more detail).

Manometer Calibration

The reading of the U-type manometer was set at zero for the condition of no flow through the test section. Liquid mercury was used as the indicator in the manometer.

Rotameter Calibration

The rotameter used for measuring liquid flow rate in the heat transfer loop was calibrated by taking known mass samples collected during a measured time interval at the operating conditions. The samples were collected at least two times for each reading. The calibration table for the rotameter is shown in Appendix A.

Heat Loss Calibration

Heat losses for the test section were measured by slowly bleeding dry saturated steam at atmospheric pressure through the system and collecting a known amount of mass of condensate for a specific time interval. The samples were collected several times during the run, starting after about eight hours of run time.

Heat losses for each run were calculated by assuming that the losses were proportional to the difference between the test section temperature and the room temperature for each as compared to the calibration run.

Loop Operating Procedure

The following step-by-step procedure was followed for each run.

1. The generator was turned on and left to warm up for about two hours with no current flow to the test section.

2. The pump was started and water was pumped through the system at the desired flow rate.

3. Cooling water was sent to the shellside of the heat exchanger.

4. After about five minutes, heat was added to the test section at the prescribed rate by adjusting the current rate to the test section.

5. After steady conditions had been reached (usually two hours steady conditions were determined by recording operating conditions at intervals of one hour and obtaining the same values for two sets), the wall temperatures, inlet and outlet temperatures, current to the test section, voltage drop, and other operating conditions such as room temperature, and atmospheric pressure were measured and recorded.

In all runs, the water bath temperature was maintained at about 75^{o}F.

CHAPTER V

EXPERIMENTAL RESULTS AND DISCUSSION

The following parametric ranges were covered during the course of study:

Reynolds Number	7,300 to 27,000
Prandtl Number	4.5 to 8.0
Mass Velocities	215,900 to 84,000 lbm/hr ft^2
Average Heat Fluxes	2,825 to 16,600 Btu/hr ft^2

A total of ten runs was made. In each run the outside wall temperatures, inlet and outlet temperatures, flow rate, current and voltage drop in the test section, room temperature, and atmospheric pressure were measured. The value of these experimental measurement is presented in Appendix B.

Local Results

The local heat transfer coefficient was calculated by the following equation

$$h = \frac{q/A}{t_{w_i} - t_B} \tag{3}$$

To compute the inside wall temperature, t_{w_i}, from the measured outside wall temperature is a very complicated problem due to the existence of peripheral temperature gradients in tube wall, nonuniform wall thickness (for the bend section) and length of conduction path, and the temperature effects on the electrical and thermal conductivities. Small corrections are also necessary for heat losses from the system. The problem is solved by an 80-element relaxation calculation at each station which uses the outside surface temperatures and total heat generation as input and gives the local inside surface temperatures, heat fluxes, q/A, and apparent heat transfer coefficients as output. This method of heat transfer coefficient calculation was developed and applied by Owhadi (6), Crain (7), Singh (8), and Farukhi (5). In this work, to find the inside wall temperature, heat flux, and heat transfer coefficient, a modified computer program originally written by Farukhi (5) was applied. A sample calculation is presented in Appendix D, and the listing of the computer program is presented in Appendix F.

For all runs, the local heat transfer coefficient at each peripheral position was calculated and is presented in Appendix E. For runs 207, 210, and 215, the local heat transfer coefficients are plotted as a function of peripheral position for all eleven stations and are shown in Figures 5 through 13. Runs 207, 210, and 215 are chosen because they represent the lowest, intermediate, and the highest Reynolds numbers obtained in this work.

An overall analysis of these figures indicates that local heat transfer coefficient is independent of peripheral position in the straight section ahead of the bend. In general for stations 4, 5, and 6 located in the bend, the heat transfer coefficients are much larger at peripheral position 5 than peripheral position 1. Since peripheral position 5 is located at the outside wall and peripheral

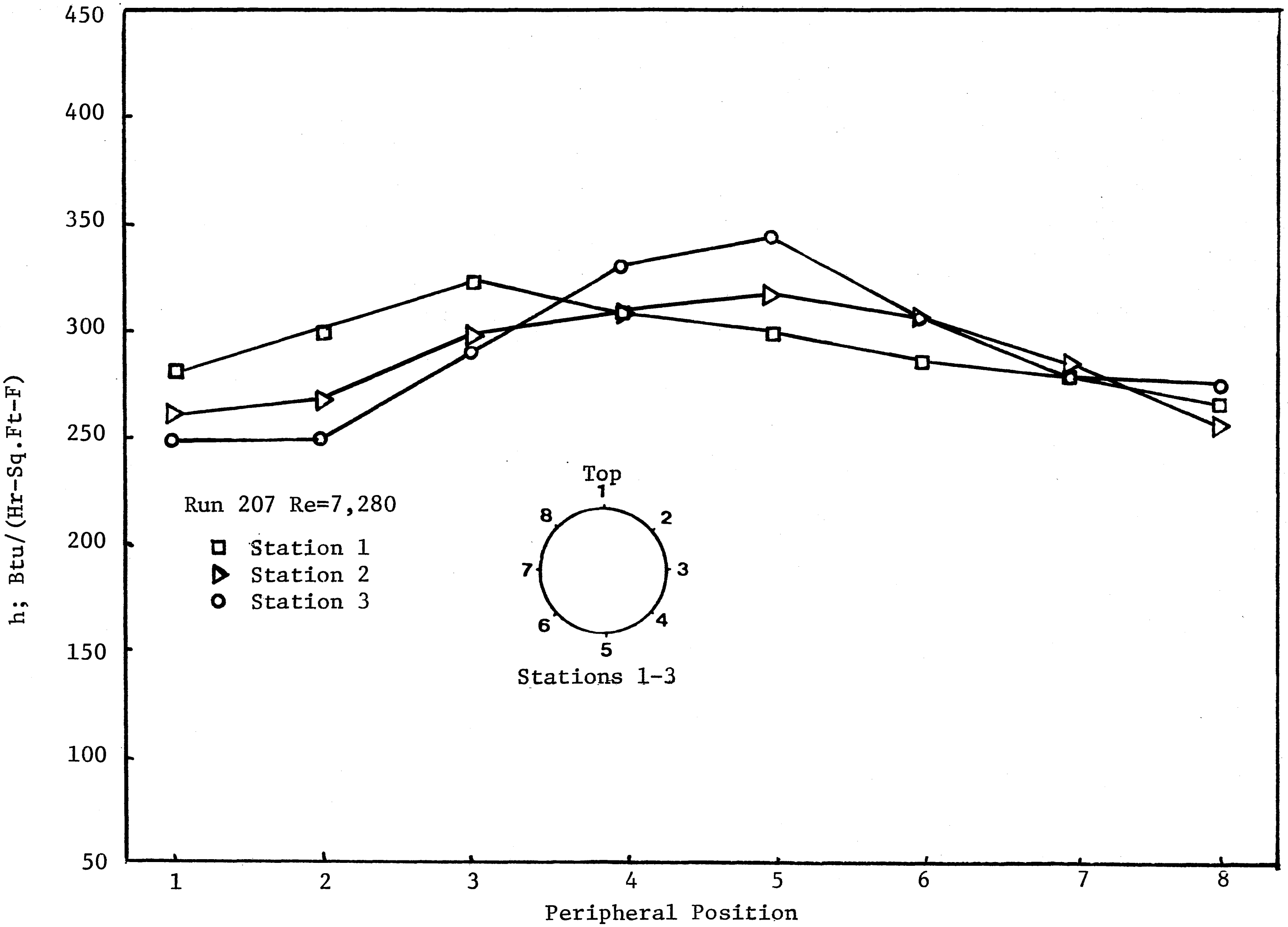


Figure 5. Peripheral Distribution of Heat Transfer Coefficients, Run 207

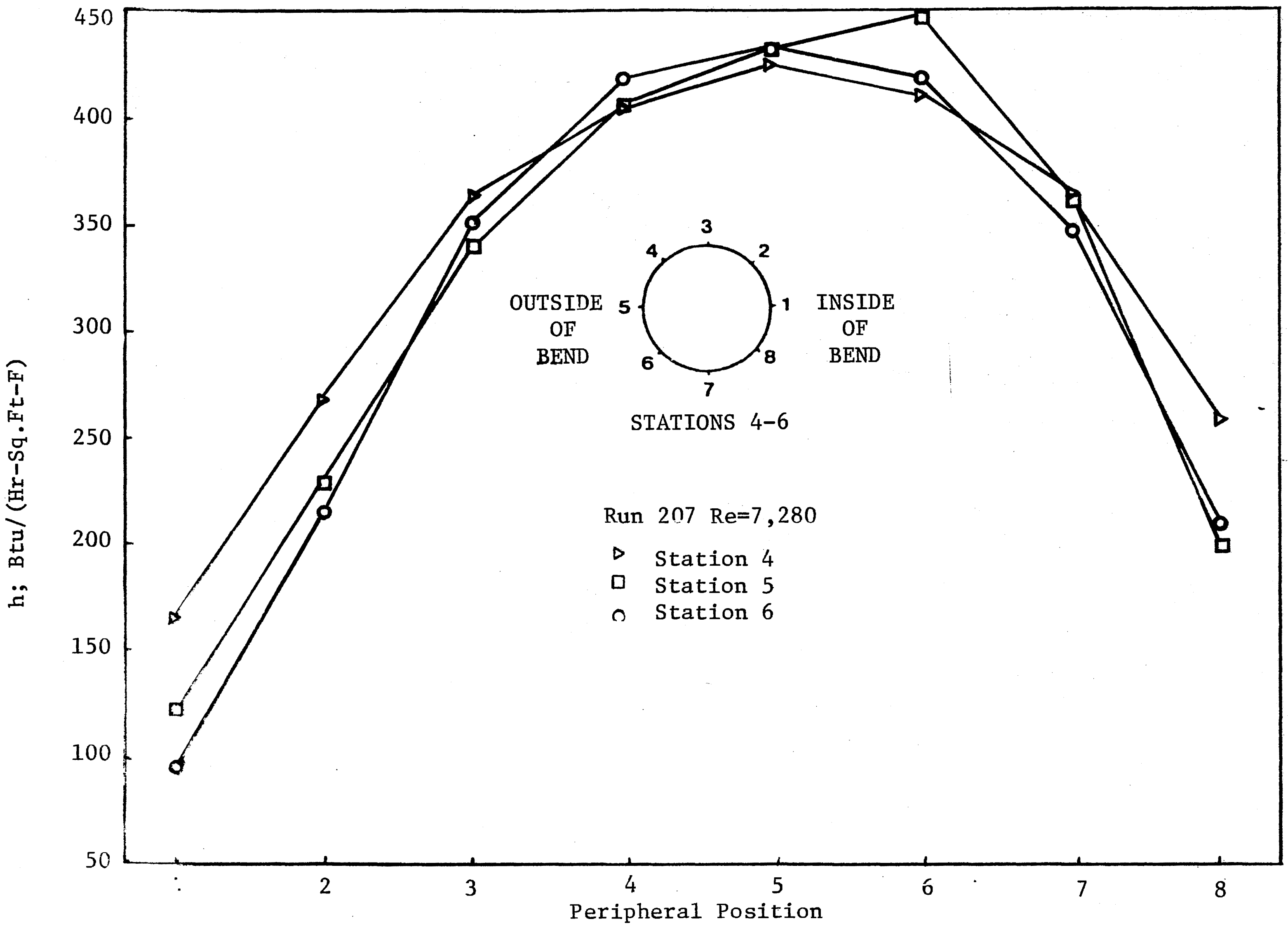


Figure 6. Peripheral Distribution of Heat Transfer Coefficients, Run 207

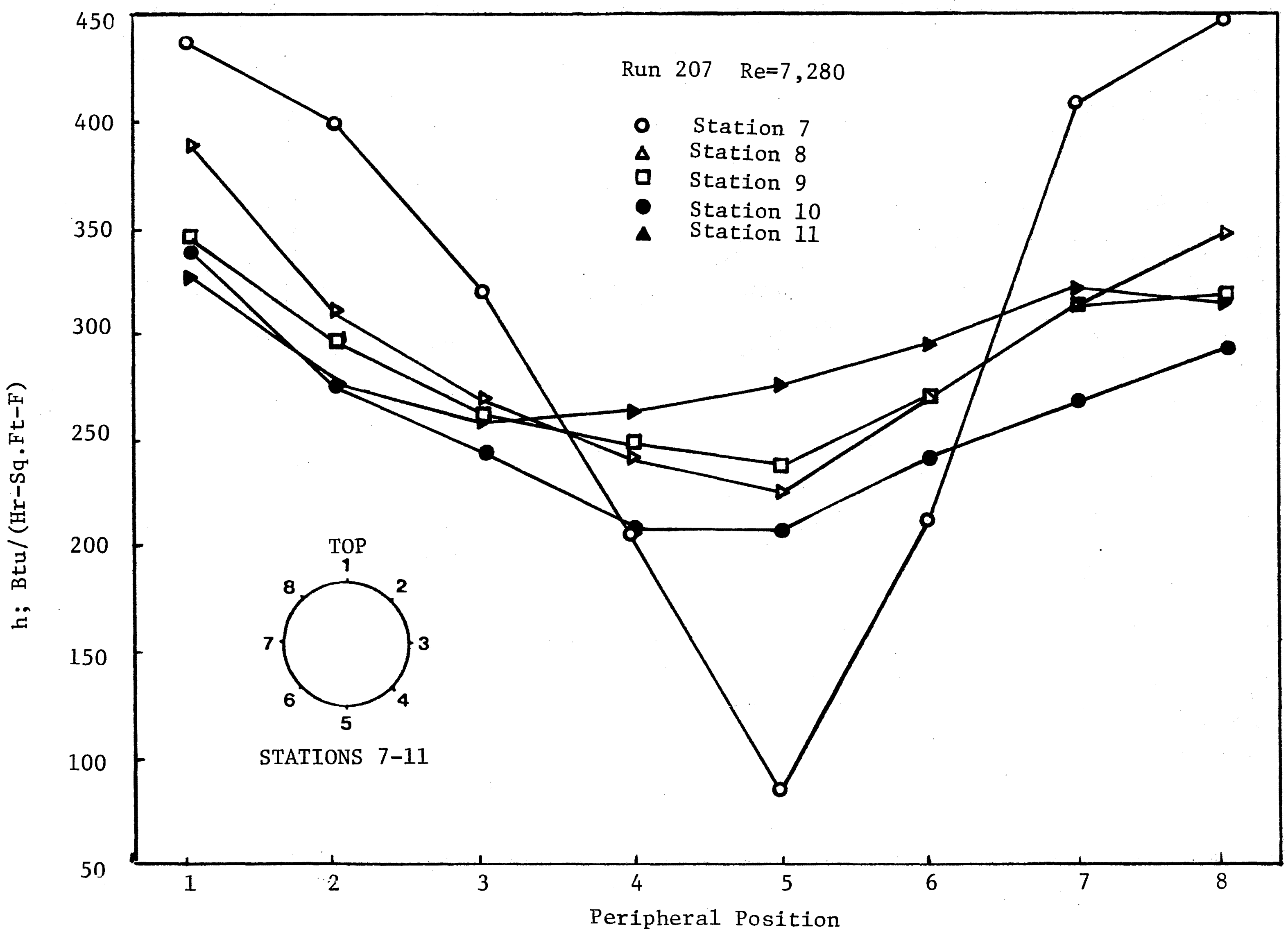


Figure 7. Peripheral Distribution of Heat Transfer Coefficients, Run 207

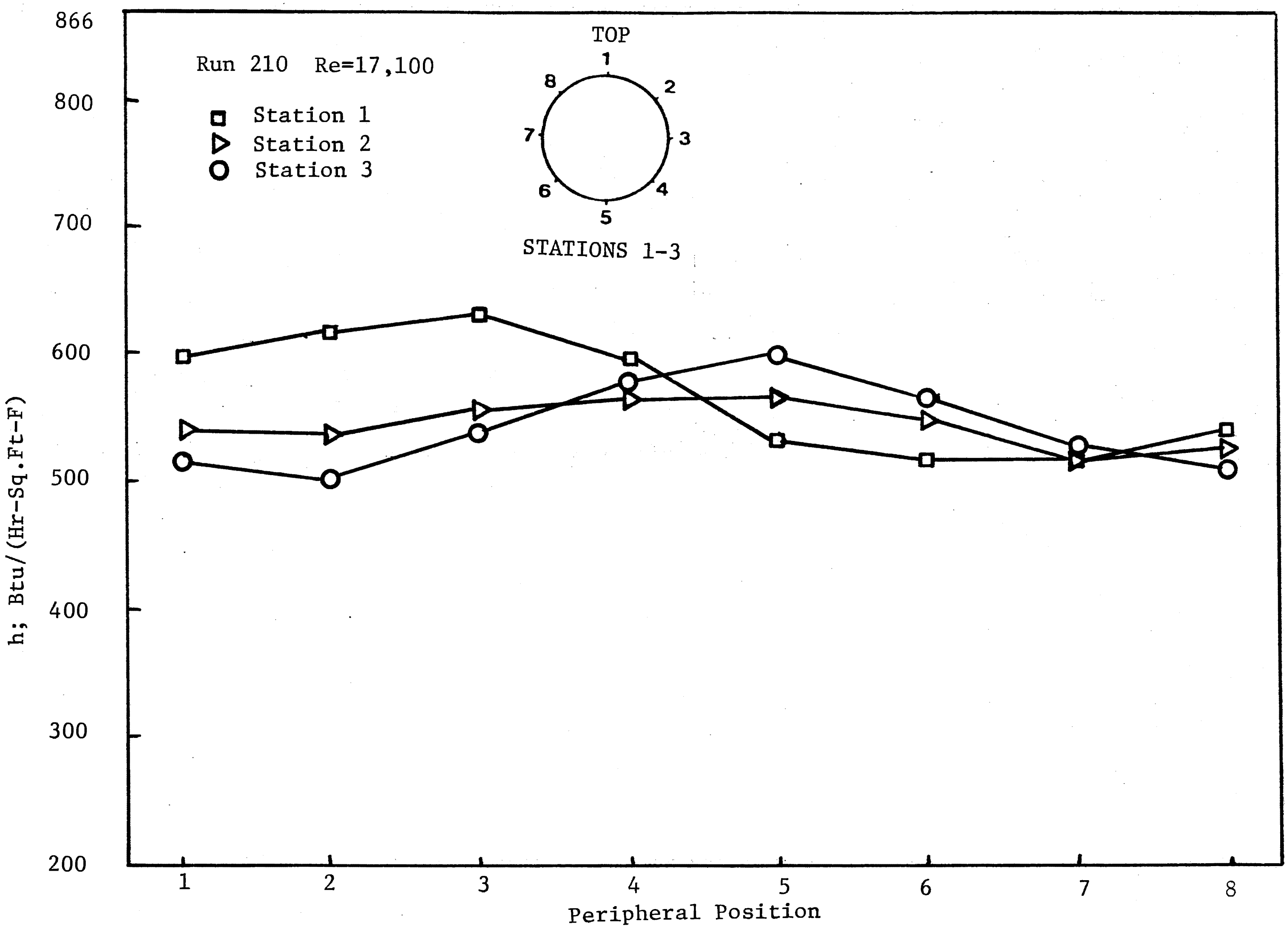


Figure 8. Peripheral Distribution of Heat Transfer Coefficients, Run 210

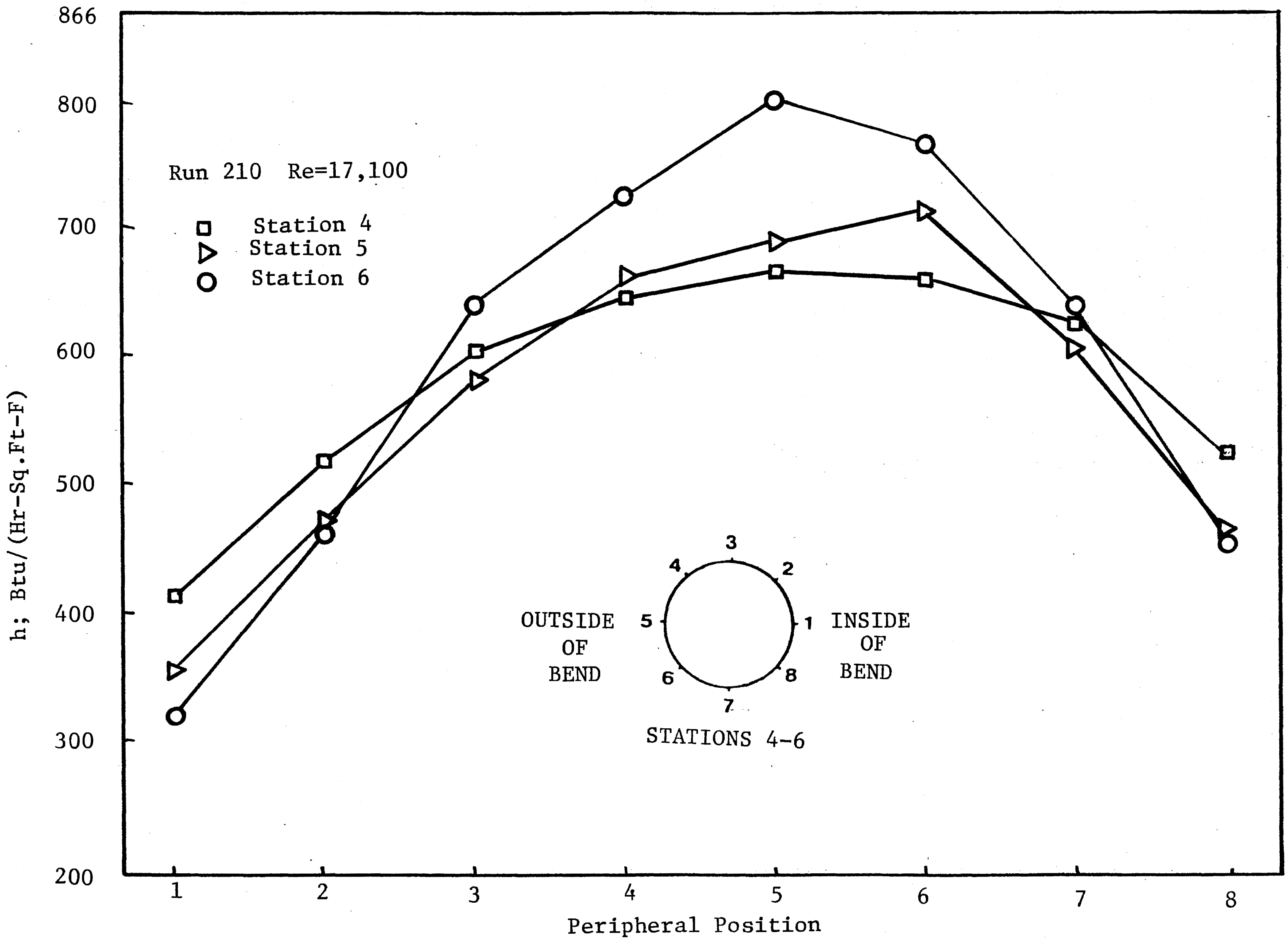


Figure 9. Peripheral Distribution of Heat Transfer Coefficients, Run 210

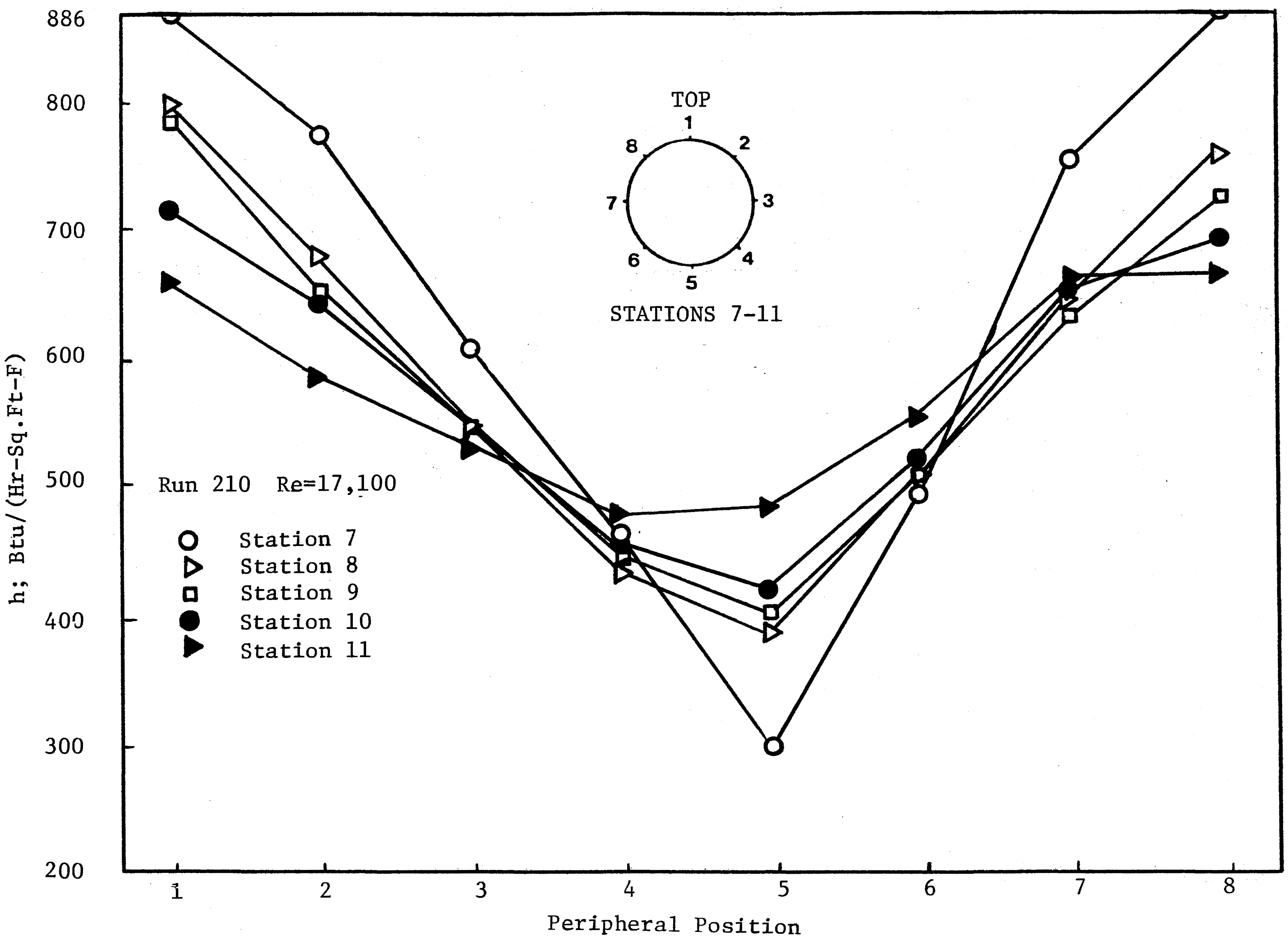


Figure 10. Peripheral Distribution of Heat Transfer Coefficients, Run 210

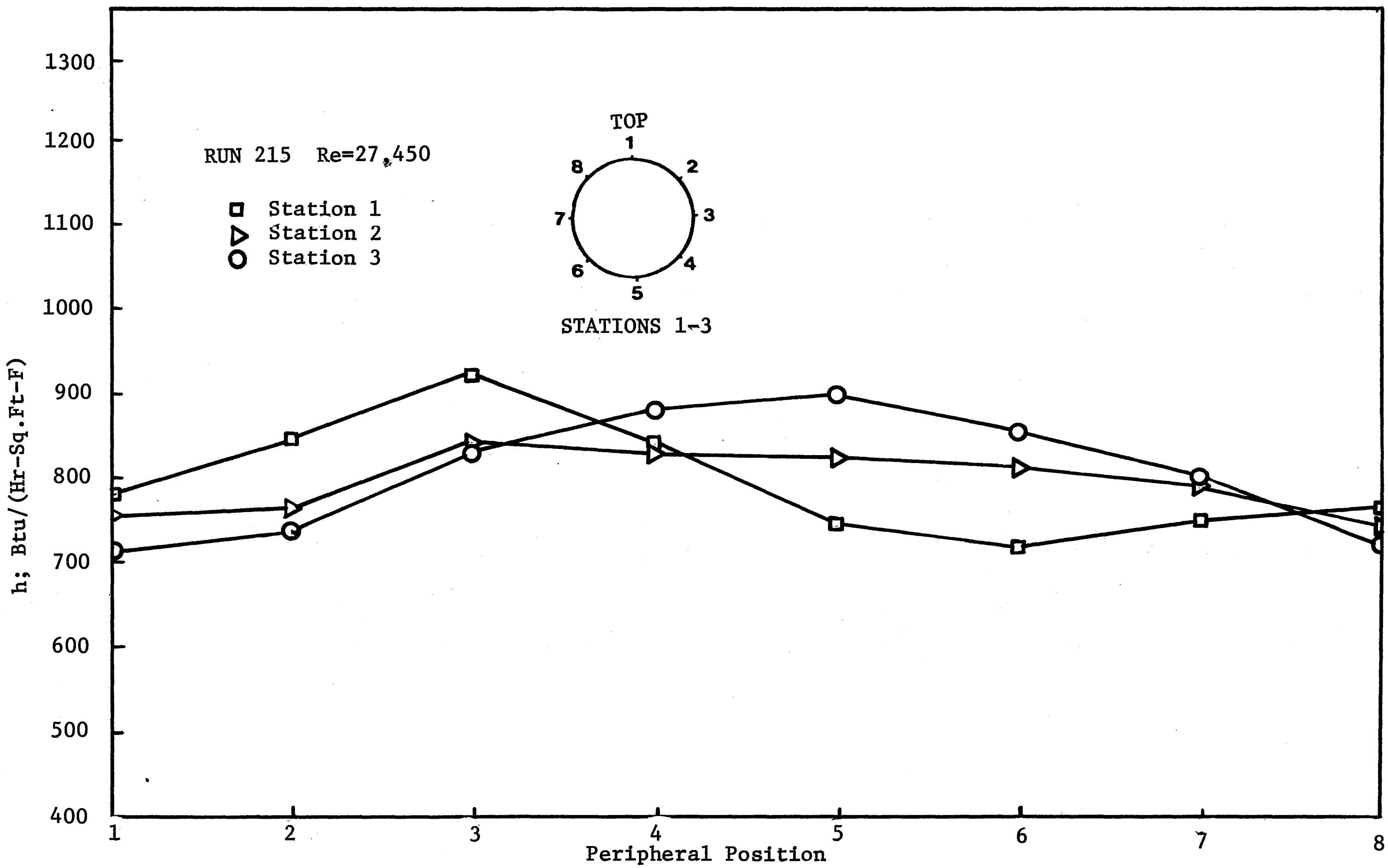


Figure 11. Peripheral Distribution of Heat Transfer Coefficients, Run 215

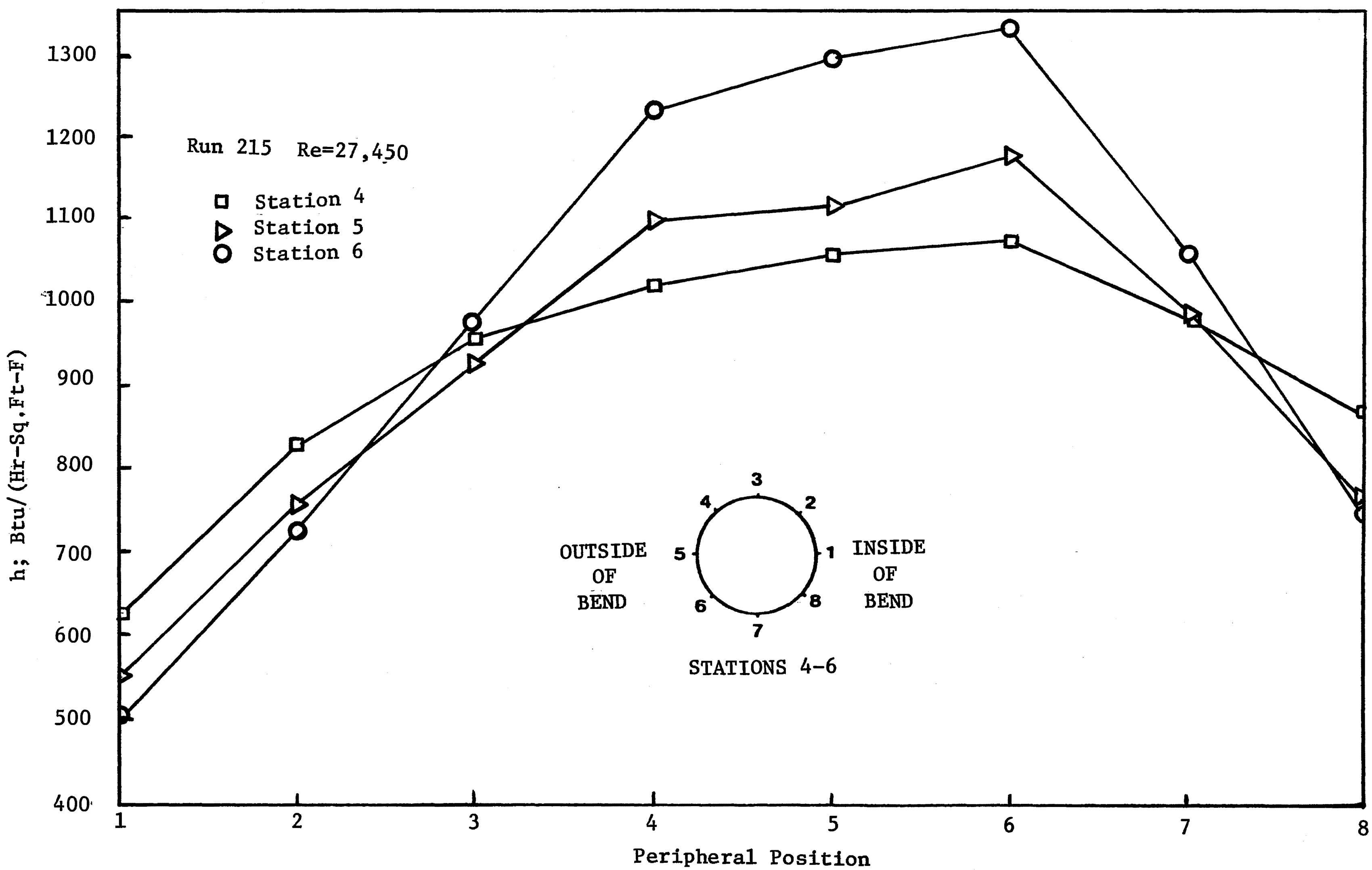


Figure 12. Peripheral Distribution of Heat Transfer Coefficients, Run 215

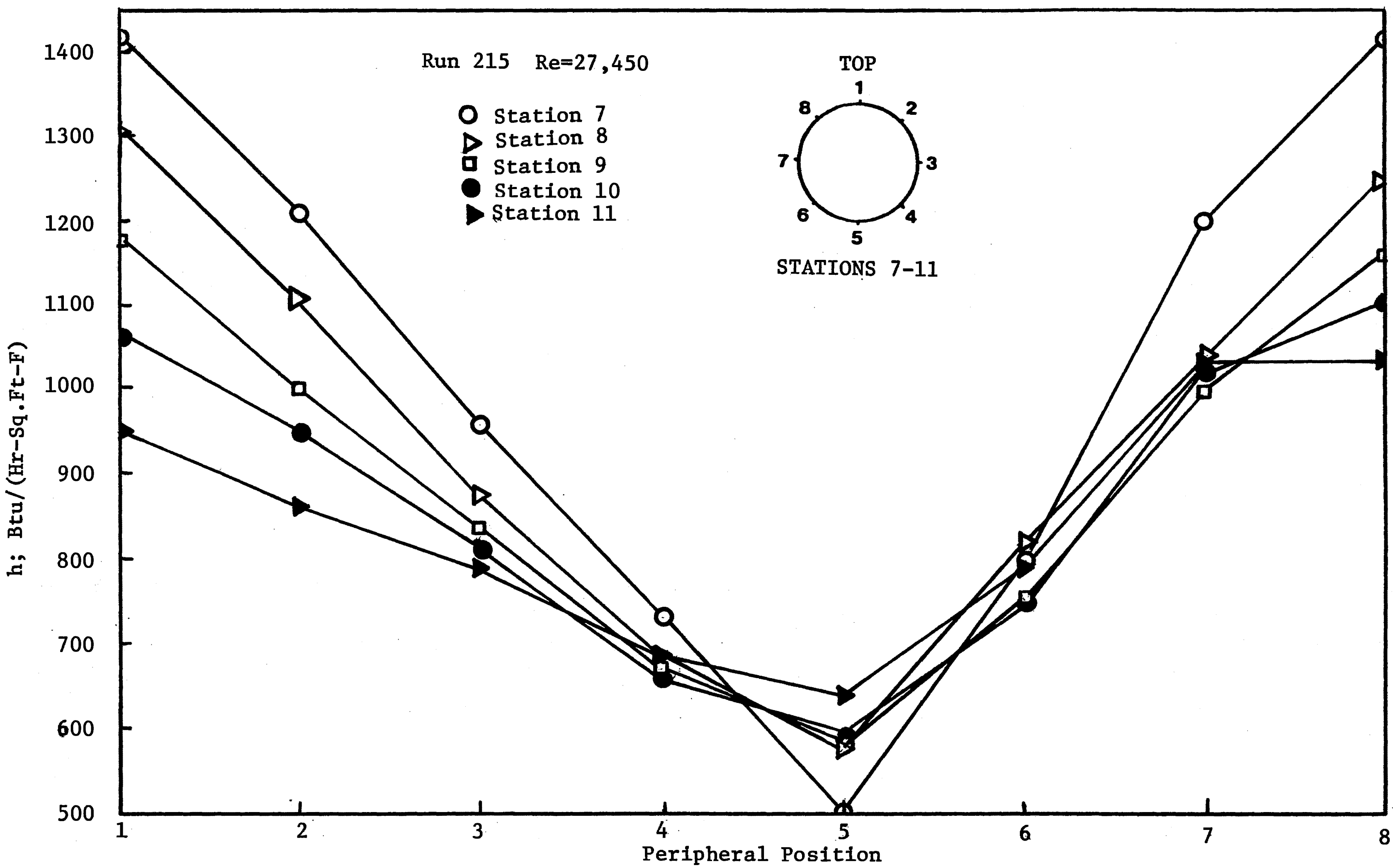


Figure 13. Peripheral Distribution of Heat Transfer Coefficients, Run 215

position 1 at the inside wall of the bend, the increase in heat transfer coefficient might be due to a higher fluid velocity at the outside wall compared to the fluid velocity near the inside wall of the bend. The higher heat transfer coefficient at outside wall of the bend might also be due to an induced secondary flow superimposed on the primary flow in the bend as shown in Figure 1. For stations downstream from the bend, the heat transfer coefficient are larger at peripheral position 1, located at the top of the tube, compared to peripheral position 5 (note that the number of peripheral position for the straight section downstream of the bend has been rotated 180°) located at the bottom of tube. The credit for higher heat transfer coefficients at the top of the tube might be given to the secondary flow caused by presence of the bend. As expected, the effect of secondary flow caused by the bend disappears at stations far downstream from the bend.

These curvature effects are in good qualitative agreement with the investigation of Ede (1) and Lis et al. (2).

Average Results

For each station the average heat transfer coefficient was calculated by the following equation

$$\bar{h} = \frac{1}{8} \sum_{i=1}^{8} (h_i) \qquad (4)$$

where h_i is the local heat transfer coefficient at each peripheral position. The calculated average heat transfer coefficient for each station is presented in Appendix E.

The calculated average heat transfer coefficients from experimental data for each station were compared to literature data. The ratios of average heat transfer coefficients from this work to heat transfer coefficients predicted by Dittus-Boelter (9) and Sieder-Tate (10) equations and heat transfer coefficients obtained from the Eagle-Ferguson (11) chart are presented in Table I.

Table I indicates that in the straight section upstream of the bend, the heat transfer coefficients found in this study are in good agreement with the Eagle-Ferguson (11) chart. Also, the heat transfer coefficients in the bend section are larger than the heat transfer coefficients obtained from Eagle-Ferguson (11) chart. For the runs made, the maximum increase of heat transfer coefficient in the bend was 18 percent, for run 211. In general this difference increases as Reynolds number increases. The bend also causes the heat transfer coefficient to increase in the straight section downstream from the 180^{o} bend. The maximum increase of heat transfer coefficient occurred at station 7, located at the entrance of the straight section downstream from the bend, and was 22 percent higher as compared to Eagle-Ferguson (11) chart, for the highest Reynolds number (Run 215) made. Again, the higher Reynolds numbers give the greater increases in heat transfer coefficients. Finally, the ratio approaches unity for stations far downstream from the bend.

The average Nusselt number, $\overline{Nu}$, for each station is calculated and plotted in Figure 14 as a function of X/d and Reynolds number. Figure 14 indicates the behavior of $\overline{Nu}$ is independent of Reynolds number for a certain region of flow, (turbulent or transition); but its value increases with Reynolds number for a given station. Figure

TABLE I

RATIO OF HEAT TRANSFER COEFFICIENTS FROM PRESENT WORK TO THOSE PREDICTED BY LITERATURE

No.	Re_D	Source		1	2	3	4	5	6	7	8	9	10	11
206	7,300	Dittus-Boelter	(D-B)	0.95	0.91	0.92	1.06	1.01	1.01	0.99	0.80	0.94	0.90	0.90
		Sieder-Tate	(S-T)	0.90	0.86	0.86	1.00	0.94	0.96	0.93	0.74	0.88	0.85	0.84
		Eagle-Ferguson	(E-F)	1.02	0.98	0.98	1.14	1.08	1.10	1.06	0.85	1.00	0.97	0.96
207	7,270	D-B		0.98	0.95	0.95	1.10	1.03	1.03	1.02	0.84	0.96	0.92	0.93
		S-T		0.92	0.89	0.89	1.03	0.96	0.96	0.96	0.79	0.90	0.86	0.87
		E-F		1.05	1.02	1.02	1.13	1.10	1.10	1.09	0.90	1.02	0.98	0.99
208	13,400	D-B		0.95	0.93	0.94	1.04	1.03	1.06	1.03	0.95	0.89	0.85	0.83
		S-T		0.91	0.88	0.90	0.99	0.98	1.01	0.98	0.91	0.85	0.81	0.79
		E-F		1.01	0.98	1.00	1.11	1.10	1.13	1.10	1.01	0.95	0.91	0.89
209	16,580	D-B		0.93	0.88	0.87	0.95	0.93	0.97	1.02	0.97	0.94	0.93	0.89
		S-T		0.86	0.81	0.80	0.88	0.85	0.90	0.94	0.89	0.87	0.85	0.82
		E-F		0.99	0.94	0.93	1.01	0.99	1.04	1.08	1.03	1.00	0.99	0.95

TABLE I (Continued)

No.	Re_D	Source	1	2	3	4	5	6	7	8	9	10	11
210	17,000	D-B	0.92	0.87	0.87	0.94	0.91	0.96	1.02	0.94	0.93	0.92	0.91
		S-T	0.88	0.83	0.82	0.89	0.86	0.91	0.97	0.90	0.89	0.87	0.86
		E-F	0.98	0.93	0.92	0.99	0.96	1.02	1.09	1.00	0.99	0.98	0.97
211	19,850	D-B	1.01	0.95	0.95	1.06	1.06	1.11	1.14	1.07	1.00	0.96	0.94
		S-T	0.94	0.87	0.87	0.98	0.97	1.02	1.05	0.98	0.92	0.88	0.86
		E-F	1.08	1.01	1.01	1.13	1.12	1.18	1.21	1.13	1.06	1.03	0.99
212	19,350	D-B	0.92	0.89	0.89	0.99	0.99	1.06	1.11	1.06	1.03	1.01	0.95
		S-T	0.88	0.85	0.85	0.94	0.94	1.00	1.05	1.01	0.98	0.96	0.91
		E-F	0.98	0.95	0.94	1.05	1.05	1.12	1.17	1.13	1.09	1.07	1.01
213	22,370	D-B	0.96	0.93	0.92	1.03	1.03	1.08	1.13	1.05	0.99	0.96	0.91
		S-T	0.89	0.86	0.85	0.95	0.95	1.00	1.04	0.97	0.91	0.88	0.83
		E-F	1.02	0.99	0.98	1.10	1.09	1.15	1.20	1.12	1.05	1.02	0.96
214	23,950	D-B	0.95	0.93	0.93	1.05	1.03	1.08	1.11	1.03	0.97	0.95	0.92
		S-T	0.89	0.87	0.86	0.98	0.96	1.01	1.03	0.95	0.91	0.89	0.85

TABLE I (Continued)

No.	Re_D	Source	1	2	3	4	5	6	7	8	9	10	11
		E-F	1.01	0.99	0.98	1.11	1.10	1.15	1.17	1.09	1.04	1.01	0.97
215	27,320	D-B	0.93	0.91	0.91	1.05	1.04	1.10	1.15	1.06	0.99	0.96	0.92
		S-T	0.86	0.84	0.85	0.98	0.96	1.02	1.07	0.99	0.92	0.89	0.85
		E-F	0.98	0.96	0.97	1.12	1.10	1.17	1.22	1.13	1.05	1.02	0.98

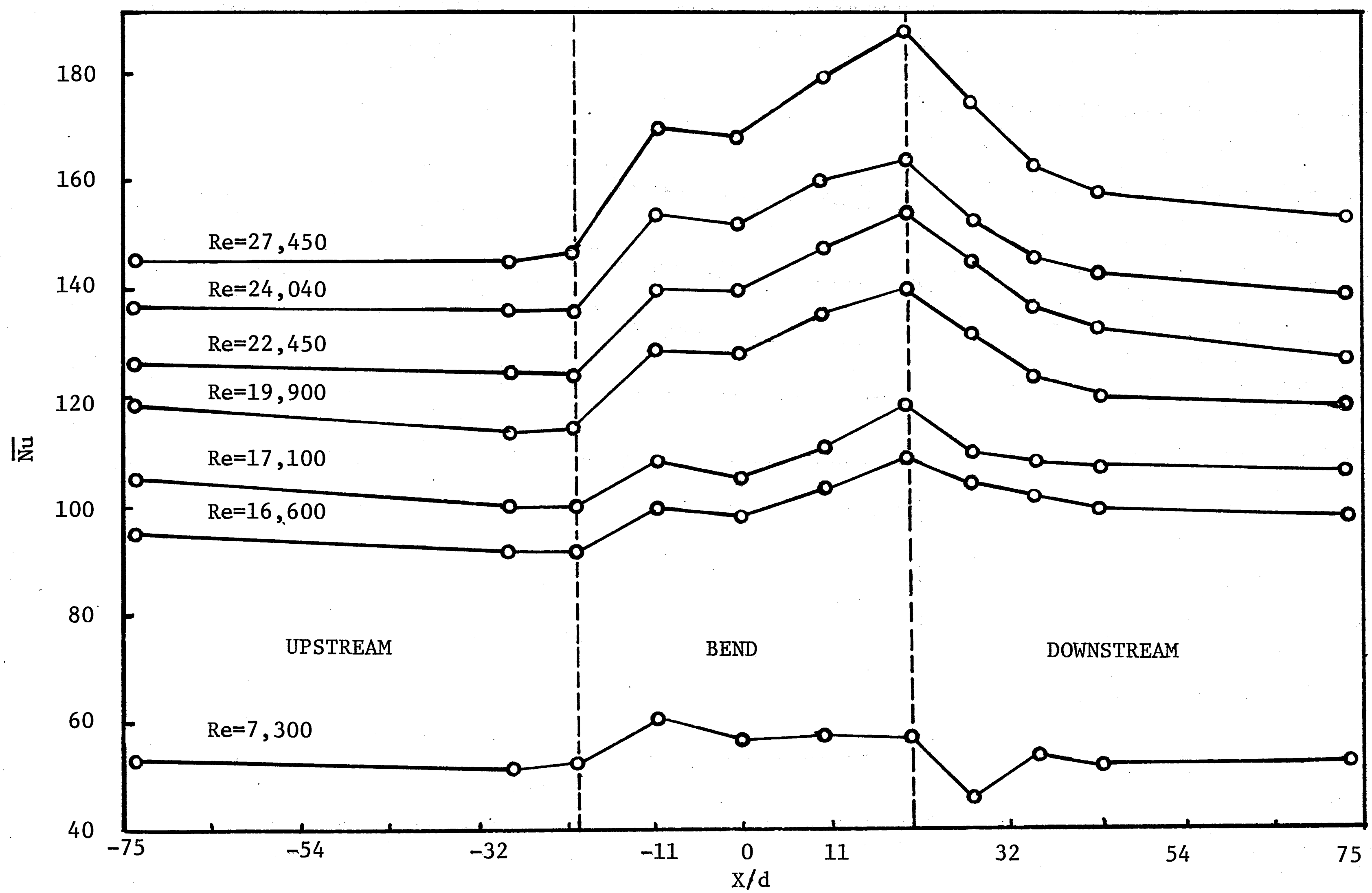


Figure 14. $\overline{Nu}$ Number as a Function of Re and Position in Test Section

14 also shows the $\overline{Nu}$ is fairly constant in the straight section upstream of the bend but its value is enhanced in the bend for both the turbulent and transition regions. For the turbulent flow, the value of $\overline{Nu}$ starts to decrease and approaches a constant value in the straight section downstream of the bend. For the same portion of the test section the behavior of $\overline{Nu}$ number is different for the transition region, as the $\overline{Nu}$ number decreases to a minimum and then goes back to a constant value.

Error Analysis

The heat balances for all ten runs were made and the results indicate a maximum of ± 5 percent error. The results of the heat balance calculation are presented in Appendix E.

The accuracy of the electrical measuring devices is reported to be within 2 percent of full scale reading. Also, the accuracy of rotameter is expected to be within 2 percent. The accuracy of inside wall temperature is expected to be within 1 percent and that of room temperature ± 0.5 percent. The accuracy of longitudinal test section dimensions is also expected to be within ± 0.1 percent.

If all of the above mentioned measurements were in error to the extent indicated and in the same direction, the maximum error in the heat flux would be about 4 percent. Based on the error analysis which was made by Singh (8) the above mentioned inaccuracy of measurements gives a maximum error of about 12 percent in the heat transfer coefficient calculation. However, the average error is about 6 percent.

CHAPTER VII

CONCLUSIONS AND RECOMMENDATION

Analysis of the experimental data obtained in this investigation of the 180° bend indicates that:

1. The local heat transfer coefficient in the straight section upstream of the bend is independent of peripheral position.

2. The peripheral heat transfer is far from uniform in the bend, being much higher at the outside and lower at the inside of the bend, resulting in a higher mean heat transfer coefficient as compared to a straight tube under similar conditions.

3. At any cross section in the bend, the distribution of peripheral heat transfer coefficient is almost symmetrical about a plane containing the longitudinal axis of tube and the radius of the bend.

4. At the entrance region of the downstream section of the bend, the distribution of peripheral heat transfer coefficient is symmetrical about the plane of the bend. Also, the peripheral heat transfer coefficient is highest at the outside wall of the bend and lowest at the inside wall. The nonuniform heat transfer coefficient results in a higher mean heat transfer coefficient as compared to a straight section under similar conditions without the bend. The nonuniform behavior of the local heat transfer coefficient disappears as fluid flows further down the tube.

No attempt was made to find a correlation(s) to predict the heat transfer coefficient in the bend and in the entrance region of the straight section downstream from the bend due to the lack of enough data. Therefore, using different bend to tube diameter ratios, different fluids and covering a wider range of Reynolds number, one could find a correlation(s) to predict the heat transfer coefficient for the bend and the entrance region of the straight section downstream of the bend.

BIBLIOGRAPHY

(1) Lis, J. and M. J. Thelwell, "Experimental Investigation of Turbulent Heat Transfer in a Pipe Preceded by a 180° Bend," Proc. Inst. of Mech. Engrs., V. 178, Pt. 3I, 1963-64, p. 17.

(2) Ede, A. J., "The Effect of a 180° Bend Heat Transfer to Water in a Tube," 3rd Intl. Heat Transfer Conf., Chicago, Ill., V. 1, 1966, p. 99.

(3) Tailby, S. R. and P. W. Staddon, "The Influence of 90° and 180° Pipe Bend on Heat Transfer from an Internally Flowing Gas Stream," 4th Intl. Heat Transfer Conf., Paris-Vesailles, V. 2, FC4.5(1970).

(4) Yang, J. W. and N. Liao, "Turbulent Heat Transfer in Rectangular Ducts with 180°-Bend," Heat Transfer, V. 2 -169, 1974.

(5) Farukhi, M. N., "An Experimental Investigation of Forced Convective Boiling at High Qualities Inside Tubes Preceded by 180 Degree Bends," Ph.D. Thesis, Oklahoma State University, Stillwater, Oklahoma (1974).

(6) Owhadi, A., K. J. Bell, and B. Crain, Jr., "Forced Convection Boiling Inside Helically Coiled Tubes," Intl. J. of Heat and Mass Transfer, V. 11, 1968, p. 1779.

(7) Crain, Berry, Jr., "Forced Convection Heat Transfer to a Two-Phase Mixture of Water and Steam in a Helical Coil," Ph.D. Thesis, Oklahoma State University, Stillwater, Oklahoma (1973).

(8) Singh, S. P., "Liquid Phase Heat Transfer in Helically Coiled Tubes," Ph.D. Thesis, Oklahoma State University, Stillwater, Oklahoma (1973).

(9) Dittus, F. W. and L. M. K. Boelter, Univ. of Calif. (Berkeley) Eng. Pub., Vo. 2, 1930, p. 443.

(10) Sieder, E. N. and C. E. Tate, "Heat Transfer and Pressure Drop of Liquids in Tubes," Ind. Eng. Chem., V. 28, 1936, p. 1429.

(11) Eagle, A. and R. M. Ferguson, Pro. Roy. Soc., A 127, 540-566(1930).

APPENDIX A

CALIBRATION DATA

TABLE II

CALIBRATION DATA FOR ROTAMETER

Rotameter Setting % Maximum Flow	Mass Flow Rate, lbm/hr
10	355.0
20	651.0
30	937.0
40	1219.0
50	1530.0
60	1835.0
70	2118.0
80	2433.0
90	2744.0
100	3013.0

TABLE III

CALIBRATION DATA FOR INLET AND OUTLET THERMOCOUPLES DURING IN-SITU CALIBRATION OF SURFACE THERMOCOUPLES ON THE TEST SECTION

Thermocouple Location	Saturated Steam Temperature, ^{o}F	Thermocouple Correction, ^{o}F	Average Room Temperature
Inlet	210.3	-0.7	76.5
Outlet	211.1	-0.9	

Therefore the correction correlation for inlet and outlet bulk fluid temperature are

$$(T_{in})_{corrected} = T_{in} - (0.7)\,\frac{(T_{in}-T_{room})}{(210.3-76.5)}$$

and

$$(T_{out})_{corrected} = T_{out} - (0.9)\,\frac{(T_{out}-T_{room})}{(211.1-76.5)}$$

TABLE IV

CALIBRATION DATA FOR HEAT LOSS FROM TEST SECTION

Average temperature of steam in test section	= $210.3^{o}F$
Room Temperature during calibration	= $76.0^{o}F$
Amount of mass of steam condensed	= 0.50 lbm/hr
Heat of vaporization of water at $210.3^{o}F$ (λ)	= 971.36 Btu/lbm

$$\left\{\begin{matrix}\text{Heat loss from} \\ \text{heat transfer loop}\end{matrix}\right\} = Q = m\lambda$$

$$= (0.5)(971.36)$$

$$= 485 \text{ Btu/hr}$$

So heat loss during experimental run, Q_{loss}-Btu/hr

$$Q_{loss} = \frac{485\ (T_{avg} - T_{room})}{(210.3-76.0)}$$

or, heat lost from the test section, q_{loss}-Btu/hr

$$q_{loss} = \left(\frac{\ell}{L_{total}}\right) Q_{loss}$$

where

ℓ = Length of test section.

and

L = Total length of heat transfer loop.

$$T_{avg} = \left(1/2\right)(T_{in} + T_{out})\ ,\ {}^{o}F$$

TABLE V

CALIBRATION DATA FOR CALIBRATION OF OUTSIDE SURFACE THERMOCOUPLES

Station Number	Δ = (Steam Temperature) - (Thermocouple Temperature) Peripheral Position							
	1	2	3	4	5	6	7	8
1	-0.2	-0.3	-0.6	+0.1	+0.3	+0.5	-0.3	-0.4
2	+0.1	-0.6	-0.5	+0.1	-0.2	-0.2	-0.6	+1.3
3	-0.1	-0.6	-0.1	-0.6	0.0	-0.1	-0.2	-0.5
4	+0.5	-0.7	-0.6	-0.6	-0.2	-0.3	-0.2	-0.5
5	-0.5	-0.7	-0.6	-0.3	-0.6	-0.4	-0.4	-0.3
6	-0.6	-0.7	-0.1	-0.4	-0.6	-0.6	-0.7	-0.6
7	-0.7	-0.8	-0.6	-0.8	-0.8	-0.7	-0.7	-0.5
8	-0.5	-0.7	-1.0	-0.8	-0.5	-0.8	-0.5	-0.7
9	-0.9	-0.5	-0.9	-0.8	-0.3	-0.6	-0.5	-0.8
10	-0.7	-1.0	-0.6	-0.9	-0.4	-0.9	-1.0	-0.8
11.	-0.6	-0.8	-0.5	-0.8	-0.6	-0.9	-1.0	-0.7

Note: 1. For correction of the surface thermocouples the effect of pressure drop along the pipe is not included even though the pressure taps indicated an average pressure drop of 0.5 in. Hg along the pipe.

2. Since the difference between the thermocouple reading and the true temperature of surface was small, no correction for surface temperatures was made through the further calculations.

APPENDIX B

EXPERIMENTAL DATA

RUN NUMBER 206

WATER FLOW RATE	=	700.00	LBM/HOUR
CURRENT TO TUBE	=	215.00	AMPS
VOLTAGE DROP IN TUBE	=	8.40	VOLTS
ROOM TEMPERATURE	=	79.90	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	83.00	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	91.90	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	94.2	97.6	98.6	104.1	109.2	112.4	96.3	99.8	98.2	99.7	101.9
2	93.6	97.4	98.4	99.1	102.2	103.2	97.0	101.3	99.2	100.3	102.9
3	93.2	96.5	97.0	95.9	97.0	97.5	98.9	102.0	100.7	101.4	103.3
4	93.4	95.9	96.0	94.4	94.9	95.4	104.0	102.9	101.9	101.9	103.3
5	93.8	96.0	95.6	93.9	94.2	94.5	112.0	103.1	102.4	102.2	102.7
6	94.2	96.3	96.6	94.3	94.6	94.8	103.3	101.4	100.9	101.2	102.1
7	94.5	97.0	97.5	95.7	96.7	97.2	97.1	100.5	99.2	99.9	101.4
8	94.6	97.6	98.0	99.3	102.9	104.1	95.7	99.8	98.3	99.5	101.4

RUN NUMBER 207

WATER FLOW RATE	=	700.00	LBM/HOUR
CURRENT TO TUBE	=	224.00	AMPS
VOLTAGE DROP IN TUBE	=	8.70	VOLTS
ROOM TEMPERATURE	=	80.20	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	82.30	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	91.80	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	94.3	97.8	98.8	104.4	109.8	113.4	96.1	99.0	98.2	99.6	101.9
2	93.7	97.5	98.6	99.5	102.6	104.2	96.9	100.6	99.8	100.8	103.2
3	93.1	96.6	97.2	95.9	97.3	97.8	99.2	102.1	101.3	102.1	103.8
4	93.4	96.1	96.0	94.4	95.0	95.5	104.8	103.8	102.5	102.7	103.8
5	93.7	95.9	95.7	93.7	94.1	94.8	113.3	103.8	103.0	103.0	103.3
6	94.1	96.2	96.6	94.3	94.8	95.1	104.1	102.2	101.3	101.8	102.6
7	94.4	97.0	97.4	95.9	97.2	97.6	97.3	100.9	99.8	100.4	101.9
8	94.5	97.5	98.4	99.8	103.4	104.9	96.0	99.9	98.8	100.0	101.9

RUN NUMBER 208

WATER FLOW RATE	= 1534.80	LBM/HOUR
CURRENT TO TUBE	= 245.00	AMPS
VOLTAGE DROP IN TUBE	= 9.50	VOLTS
ROOM TEMPERATURE	= 79.40	DEGREES F
UNCORRECTED INLET TEMPERATURE	= 70.20	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	= 75.60	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	78.6	80.6	81.1	84.0	86.6	88.3	79.4	80.2	81.1	82.1	83.7
2	78.2	80.6	81.1	81.9	83.3	83.9	80.0	81.0	82.1	82.9	84.7
3	78.2	80.3	80.6	80.0	80.4	80.7	81.5	82.8	83.4	84.0	85.3
4	78.7	80.1	80.0	79.1	79.0	79.1	84.2	84.7	84.8	85.2	85.8
5	79.0	80.1	79.8	78.5	78.4	78.5	88.2	85.6	85.5	85.6	85.6
6	79.3	80.4	80.3	78.8	78.7	78.9	83.8	83.4	83.9	84.2	84.7
7	79.2	80.7	80.9	79.9	80.3	80.2	80.6	81.8	82.5	82.8	83.9
8	79.1	80.9	81.1	81.9	83.6	84.0	79.6	80.6	81.5	82.2	83.7

RUN NUMBER 209

WATER FLOW RATE	=	1534.80	LBM/HOUR
CURRENT TO TUBE	=	412.00	AMPS
VOLTAGE DROP IN TUBE	=	16.20	VOLTS
ROOM TEMPERATURE	=	82.10	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	82.60	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	98.20	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	105.6	111.4	113.3	120.5	126.2	129.4	107.8	109.3	111.0	112.5	117.0
2	104.4	111.2	113.3	114.9	118.5	119.6	109.6	111.6	113.1	114.6	119.1
3	103.9	110.2	111.6	110.2	111.9	111.6	113.2	115.6	116.6	117.5	120.9
4	105.0	109.7	110.4	107.8	108.0	107.9	119.6	120.5	120.1	120.9	122.4
5	106.1	110.3	109.5	106.6	106.6	106.3	127.9	123.0	122.4	122.3	122.6
6	106.8	110.3	110.6	107.3	107.1	107.3	117.6	117.1	117.9	118.4	119.8
7	106.9	111.5	112.2	109.8	111.3	111.2	110.3	112.6	113.8	114.1	117.7
8	106.6	111.5	113.0	114.9	118.8	119.9	107.7	109.7	111.3	112.9	117.1

RUN NUMBER 210

WATER FLOW RATE	=	1832.10	LBM/HOUR
CURRENT TO TUBE	=	235.00	AMPS
VOLTAGE DROP IN TUBE	=	9.10	VOLTS
ROOM TEMPERATURE	=	79.90	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	75.90	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	80.10	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	82.6	84.4	84.9	87.2	88.9	89.8	83.3	83.9	84.2	84.8	86.0
2	82.4	84.4	85.0	85.5	86.6	86.9	83.8	84.6	85.0	85.3	86.6
3	82.3	84.2	84.6	84.2	84.7	84.5	85.0	85.8	86.0	86.2	87.2
4	82.6	84.1	84.2	83.5	83.6	83.5	86.8	87.3	87.3	87.3	87.8
5	83.2	84.1	84.0	83.1	83.1	82.9	89.5	88.0	87.9	87.8	87.7
6	83.4	84.3	84.3	83.4	83.3	83.3	86.3	86.3	86.5	86.5	86.9
7	83.3	84.6	84.7	84.0	84.5	84.5	84.0	84.9	85.2	85.3	86.0
8	83.1	84.5	84.9	85.4	86.6	87.0	83.3	84.1	84.5	84.9	85.9

RUN NUMBER 211

WATER FLOW RATE	=	1832.10	LBM/HOUR
CURRENT TO TUBE	=	482.00	AMPS
VOLTAGE DROP IN TUBE	=	18.80	VOLTS
ROOM TEMPERATURE	=	83.00	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	81.80	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	99.50	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	107.8	113.8	115.8	123.7	129.4	132.8	109.7	111.6	113.9	116.2	121.7
2	106.5	113.9	115.7	117.7	121.4	122.8	111.9	114.3	116.8	118.8	124.1
3	102.6	113.3	114.3	112.5	113.9	114.2	116.1	119.0	120.7	122.2	128.6
4	108.1	113.0	113.0	109.7	109.6	109.7	122.6	124.5	125.3	126.3	130.0
5	109.6	112.8	112.1	108.1	107.9	107.8	131.8	127.6	128.0	128.3	125.4
6	110.5	113.5	113.7	109.1	108.8	108.8	120.6	120.4	122.4	123.0	122.0
7	110.1	114.5	115.2	111.9	113.1	113.1	112.6	115.5	117.4	118.1	122.0
8	109.5	114.7	116.0	117.9	121.6	122.7	109.9	112.3	114.5	116.4	121.2

RUN NUMBER 212

WATER FLOW RATE	=	2129.40	LBM/HOUR
CURRENT TO TUBE	=	265.00	AMPS
VOLTAGE DROP IN TUBE	=	10.40	VOLTS
ROOM TEMPERATURE	=	79.90	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	73.70	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	78.40	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	81.7	83.5	84.1	86.4	88.2	89.1	81.8	82.3	82.7	83.3	84.8
2	81.3	83.4	83.9	84.4	85.4	85.8	82.5	83.0	83.6	84.0	85.6
3	81.0	83.0	83.4	82.9	83.2	83.2	83.7	84.3	84.6	85.0	86.3
4	81.5	83.0	83.0	82.1	82.0	81.9	85.6	86.1	86.0	86.3	87.0
5	82.0	83.0	82.8	81.7	81.5	81.3	88.6	86.9	86.9	87.0	87.0
6	82.3	83.1	83.2	81.9	81.7	81.6	85.0	84.9	85.3	85.3	86.0
7	82.2	83.4	83.6	82.7	83.0	82.9	82.6	83.4	83.8	83.8	85.0
8	82.0	83.6	84.1	84.3	85.5	85.8	81.8	82.3	82.9	83.3	84.8

RUN NUMBER 213

WATER FLOW RATE	=	2129.40	LBM/HOUR
CURRENT TO TUBE	=	496.00	AMPS
VOLTAGE DROP IN TUBE	=	19.50	VOLTS
ROOM TEMPERATURE	=	80.50	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	79.80	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	96.20	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	105.3	110.9	113.3	120.3	125.4	128.5	106.8	108.6	110.6	112.6	117.8
2	104.1	110.9	112.3	114.5	117.8	119.1	108.9	111.0	113.2	115.1	120.2
3	103.6	110.1	111.3	109.7	111.1	111.2	112.7	115.5	117.1	118.6	122.5
4	105.3	110.0	110.2	107.2	107.0	106.9	118.7	120.8	121.7	122.9	125.1
5	107.1	110.2	109.6	105.5	105.3	105.0	127.5	123.8	124.6	125.0	125.8
6	108.1	110.7	111.0	106.6	106.0	106.0	116.7	117.0	119.0	119.6	121.9
7	107.4	111.5	112.4	109.2	110.2	110.2	109.3	112.0	113.8	114.6	118.0
8	106.9	111.5	113.1	114.6	118.0	119.2	106.7	109.2	111.0	112.7	117.3

RUN NUMBER 214

WATER FLOW RATE	=	2426.70	LBM/HOUR
CURRENT TO TUBE	=	459.00	AMPS
VOLTAGE DROP IN TUBE	=	18.00	VOLTS
ROOM TEMPERATURE	=	79.50	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	76.70	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	88.70	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	97.9	102.1	103.9	109.2	113.7	115.7	97.9	99.7	101.3	102.8	106.6
2	96.6	102.0	103.3	103.8	106.3	107.6	99.5	101.6	103.2	104.3	108.1
3	95.6	100.6	101.4	100.0	101.2	101.3	102.7	105.0	106.0	106.9	109.6
4	96.8	100.4	100.5	98.0	97.9	97.9	107.5	109.0	109.6	110.4	111.7
5	98.5	100.3	99.8	96.8	96.6	96.3	114.4	111.3	112.0	112.1	112.4
6	99.0	100.7	100.8	97.5	97.1	97.2	105.9	105.9	107.4	107.7	109.1
7	98.6	101.3	101.9	99.4	100.5	100.6	100.2	102.2	103.3	103.6	106.1
8	98.6	101.9	103.2	103.7	106.6	107.7	98.2	100.0	101.4	102.4	105.8

RUN NUMBER 215

WATER FLOW RATE	=	2724.00	LBM/HOUR
CURRENT TO TUBE	=	520.00	AMPS
VOLTAGE DROP IN TUBE	=	20.50	VOLTS
ROOM TEMPERATURE	=	83.40	DEGREES F
UNCORRECTED INLET TEMPERATURE	=	77.10	DEGREES F
UNCORRECTED OUTLET TEMPERATURE	=	91.00	DEGREES F

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	2	3	4	5	6	7	8	9	10	11
1	102.8	107.6	109.5	115.4	120.6	123.2	101.6	103.4	105.4	107.5	111.9
2	101.2	107.3	108.7	108.6	111.8	113.3	103.6	105.6	107.8	109.4	113.7
3	99.8	105.5	106.4	104.5	105.9	106.0	107.2	109.6	111.1	112.4	115.5
4	101.4	105.6	105.2	102.3	102.0	101.6	112.7	114.6	115.8	116.9	118.6
5	103.7	105.7	104.8	100.9	100.6	100.1	120.8	117.6	119.0	119.4	119.9
6	104.5	106.0	105.7	101.6	101.1	100.7	110.8	111.0	113.2	113.8	115.4
7	103.7	106.5	107.0	104.0	104.9	104.8	103.9	106.5	108.0	108.3	111.0
8	103.8	107.2	108.8	107.8	111.9	113.0	101.6	103.8	105.6	106.9	110.8

APPENDIX C

REGRESSION CORRELATIONS FOR ESTIMATING THE PHYSICAL PROPERTIES OF WATER AND INCONEL 600

WATER

The following physical correlations which were developed by Singh (8) based on the CRC Handbook of Chemistry and Physics were used to compute the physical properties of water.

Density in gm/ml:

$$\rho = 0.999986 + (0.1890)(10^{-4})(T,\ ^{o}C)-(0.5886)(10^{-5})(T,\ ^{o}C)^{2} + (0.1548)(10^{-7})(T,\ ^{o}C)^{3}$$

Range: ^{o}C to $100^{o}C$

Viscosity in centipoise:

$$\log_{10} \frac{\mu_T}{\mu_{20}} = \frac{1.3272(20-T)-0.001053(T-20)^2}{T + 105}$$

where

T = water temperature, ^{o}C

and

μ_{20} = viscosity of water at $20^{o}C$ = 1.002 cp.

Range: $20^{o}C$ to $100^{o}C$

Specific heat in Btu/lb-^{o}F:

$$C_p = 1.01881-(0.4802)(10^{-3})(T,\ ^{o}F)+(0.3274)(10^{-5})(T,\ ^{o}F)^{2} -(0.604)(10^{-8})(T,\ ^{o}F)^{3}$$

Range: $32^{o}F$ to $212^{o}F$

Thermal conductivity in Btu/hr-ft-^{o}F:

$$K = 0.30289+(0.7029)(10^{-3})(T,\ ^{o}F)-(0.1178)(10^{-5})(T,\ ^{o}F)^2$$
$$-\ (0.550)(10^{-9})(T,\ ^{o}F)^3$$

Range: 44^{o}F to 206^{o}F

INCONEL 600

The following physical correlations which were developed by Farukhi (5) were used to compute the physical properties of Inconel 600.

Electrical resistivity in Ohms-in^2/in:

$$\rho = (40.40292+2.515385x10^{-3}xT)(10^{-6})$$

T = temperature, ^{o}F

Thermal conductivity in Btu/(hr-ft-^{o}F):

$$K = 0.8313769+(3.846154)(10^{-3})(T,\ ^{o}F)$$

APPENDIX D

SAMPLE CALCULATIONS

Calculations for experimental data run 209 are presented as a sample calculation. The experimental data values for run 209 are presented on page 45 in Appendix B. The sample calculations given here are based on the following assumptions and conditions:

1. Electrical resistivity and thermal conductivity of tube walls are function of temperature.

2. Peripheral wall conduction exists.

3. Axial conduction is negligible.

4. Steady state conditions exist.

5. Heat losses to the atmosphere are present.

A computer program written originally by Farukhi (5) was modified to perform the calculations for all the data runs on IBM 360/65 computer.

Calculation of the Overall Heat Balance

Heat input rate, Btu/hr = Q_{input}

$$Q_{input} = \left\{ \begin{matrix} \text{current} \\ \text{in tube} \end{matrix} \right\} \left\{ \begin{matrix} \text{voltage drop} \\ \text{across tube} \end{matrix} \right\} (3.41213)$$

$$= (412.0)(16.2)(3.41213)$$

$$= 22{,}773.9 \text{ Btu/hr.}$$

Heat output rate, Btu/hr = Q_{output}

$$Q_{output} = (W)(C_p)[(t_b)_{out} - (t_b)_{in}]$$

The inlet and outlet bulk water temperatures measured by the thermocouples were corrected, based on their calibration. Calibration data for these thermocouples are given in Table III in Appendix A.

$$\left\{\begin{matrix}\text{corrected inlet} \\ \text{water temperature}\end{matrix}\right\} = 82.6 - \frac{0.7(82.6-82.10)}{210.3-76.5}$$

$$= 82.6^{o}F$$

$$\left\{\begin{matrix}\text{corrected outlet} \\ \text{water temperature}\end{matrix}\right\} = 98.2 - \frac{0.9(98.2-82.1)}{211.1-76.5}$$

$$= 98.09^{o}F$$

$$\left\{\begin{matrix}\text{average water temperature} \\ \text{in the tube } (T_{avg})\end{matrix}\right\} = 0.5(98.09+82.6) = 90.34^{o}F$$

From Appendix C,

$$C_p = 1.01881-(0.4802)(10^{-3})(T)+(0.3274)(10^{-5})(T^2)$$
$$-(0.604)(10^{-8})(T^3)$$

at $T = 90.35^{o}F$

$$C_p = 1.01881-(0.4802)(10^{-3})(90.34)+(0.3274)(10^{-5})(90.34^2)$$
$$-(0.604)(10^{-8})(90.34^3)$$
$$= 0.998 \text{ Btu/lb-}^{o}F$$

$$Q_{output} = (1534.8)(0.998)(98.09-82.6)$$
$$= 23{,}726.0 \text{ Btu/hr}$$

Heat loss rate, Btu/hr = Q_{loss}

Calibration data for heat losses to the atmosphere are given in Table IV, Appendix A.

$$Q_{loss} = \frac{485(90.34-82.1)}{(210.3-76.0)} \times \frac{113.267}{166.767}$$

$$= 20.2 \text{ Btu/hr}$$

$$\begin{Bmatrix} \text{Percent error in} \\ \text{heat balance} \end{Bmatrix} = \frac{(Q_{input}-Q_{loss}) - Q_{output}}{0.5\left[(Q_{input}-Q_{loss}) + Q_{output}\right]} (100)$$

$$= \frac{22{,}773.9-20.2-23{,}726.0}{0.5(22{,}773.9-20.2+23{,}726.0)} (100)$$

$$= -4.18\%$$

Calculation of the Local Inside Wall Temperature
and the Inside Wall Radial Heat Flux

As indicated in Chapter V, a numerical solution developed and applied by Owhadi (6), Crain (7), Singh (8), and Farukhi (5) was used to compute the inside wall temperatures from the measured outside wall temperatures and the inside wall radial heat flux at each thermocouple location. The trial-and-error solution is complex and hence a sample calculation will not be given here; however, the derivation of equations are given in Appendix B of Farukhi's (5) thesis. Tables VI to IX give the outside surface temperatures, the computed inside wall temperatures and the inside wall radial heat fluxes for every thermocouple located on the test section.

TABLE VI

RUN @)(- OUTSIDE SURFACE TEMPERATURES, ^{o}F

Peripheral Position	Station Number 1	2	3	4	5	6	7	8	9	10	11
1	105.6	111.4	113.3	120.5	126.2	129.4	107.8	109.3	111.0	112.5	117.0
2	104.4	111.2	113.3	114.9	118.5	119.6	109.6	111.6	113.1	114.6	119.1
3	103.9	110.2	111.6	110.2	111.9	111.6	113.2	115.6	116.6	117.5	120.9
4	105.0	109.7	110.4	107.8	108.0	107.9	119.6	120.5	120.1	120.9	122.4
5	106.1	110.3	109.5	106.6	106.6	106.3	127.9	123.0	122.4	122.3	122.6
6	106.8	110.3	110.6	107.3	107.1	107.3	117.6	117.1	117.9	118.4	119.8
7	106.9	111.5	112.2	109.8	111.3	111.2	110.3	112.6	113.8	114.1	117.7
8	106.6	111.5	113.0	114.9	118.8	119.9	107.7	109.7	111.3	112.9	117.1

TABLE VII

RUN 209 - CALCULATED INSIDE SURFACE TEMPERATURE, ^{o}F

Peripheral Position	Station Number 1	2	3	4	5	6	7	8	9	10	11
1	103.1	108.9	110.8	117.8	123.5	126.8	105.2	106.7	108.4	109.9	114.4
2	101.9	108.7	110.8	112.2	115.8	116.9	107.4	109.1	110.6	112.1	116.6
3	101.3	107.7	110.1	107.6	109.4	109.0	110.6	113.1	114.1	114.9	118.4
4	102.5	107.2	107.9	105.4	105.6	105.5	117.1	118.0	117.6	118.4	119.9
5	103.6	107.8	106.9	104.3	104.4	104.0	125.6	120.6	119.9	119.9	120.1
6	104.3	107.8	108.1	104.9	104.7	104.9	115.0	114.6	115.4	115.9	117.3
7	104.4	109.0	109.7	107.2	108.7	108.6	107.7	110.1	111.3	111.5	115.1
8	104.1	109.0	110.5	112.2	116.1	117.2	105.1	107.1	108.7	110.4	114.6

TABLE VIII

RUN 209 - RADIAL HEAT FLUX FOR INSIDE SURFACE, BTU/HR-FT^2

Peripheral Position	Station Number 1	2	3	4	5	6	7	8	9	10	11
1	10403.3	10412.1	10400.7	10915.2	10902.5	10441.7	10518.7	10574.8	10558.3	10564.7	10548.4
2	10454.4	10372.2	10321.4	11004.7	11118.7	11059.9	10523.9	10517.3	10501.1	10467.4	10405.9
3	10505.9	10445.8	10446.8	10516.5	10521.7	10631.3	10577.8	10470.6	10420.4	10449.5	10405.8
4	10415.1	10480.3	10435.2	9936.0	9893.6	9988.8	10528.9	10283.3	10351.5	10306.1	10348.0
5	10392.6	10383.7	10532.1	9483.4	9276.4	9525.3	9355.7	9939.9	10032.2	10118.2	10251.5
6	10380.2	10486.1	10447.0	9969.7	9960.4	10034.2	10591.7	10500.3	10443.2	10398.0	10463.0
7	10392.0	10349.4	10373.3	10533.3	10556.8	10660.1	10686.6	10511.1	10512.2	10597.5	10509.0
8	10375.6	10411.9	10389.5	10981.9	11049.8	11001.9	10575.0	10563.9	10546.9	10467.0	10452.1

Calculation of the local heat transfer coefficient for thermocouple 1-1 (station 1, peripheral position 1):

$$\begin{Bmatrix}\text{local heat transfer} \\ \text{coefficient (h)}\end{Bmatrix} = \frac{(q/A)}{[(t_w)_1 - (t_b)_1]}$$

$$(t_b)_1 = 82.6+(98.09-82.6)\left(\frac{4.0}{113.276}\right)$$

$$= 83.1^{\circ}F$$

$$= \frac{10403.3}{103.1-83.1} = 520.2 \text{ Btu/hr-ft}^2\text{-}^{\circ}F$$

Calculation of the Relevant Dimensionless Numbers

Reynolds Number: Re

$$Re = \frac{(d_i)(G)}{(\mu)}$$

From Appendix C,

$$\log_{10}\left(\frac{\mu_T}{\mu_{20}}\right) = \frac{1.3272(20-T)-0.001053(T-20)^2}{T + 105}$$

where

T measured in $^{\circ}C$ and $\mu_{20} = 1.002$ cp.

at $(t_b)_1 = 83.1^{\circ}F = 28.39^{\circ}C$

$$\log_{10}\left(\frac{\mu_T}{1.002}\right) = \frac{1.3272(20-28.39)-0.001053(28.39-20)^2}{28.39 + 105}$$

$$\mu_T = (1.002)(0.8241) = 0.8257 \text{ cp}$$

or

$$\mu_T = (0.8257)(2.42) = 1.9983 \text{ lbm/ft-hr.}$$

Similarly for average inside surface temperature, $(T_w)_{avg}$:

$$(T_w)_{avg} = \frac{1}{8}\sum_{i=1}^{8} (T_{w_i})$$

$$= 1/8(103.1+101.9+101.3+102.5+103.6+104.3+104.4+104.1)$$

$$= 103.15^{\circ}F$$

$$= 39.52^{\circ}C$$

So at $(T_w)_{avg} = 39.52$: $\mu_w = 0.6589$ cp

$$= 1.5946 \text{ lbm/ft-hr}$$

$$G = \frac{W}{\frac{\pi}{4}(d_i)^2} = \frac{1534.8}{\frac{\pi}{4}(\frac{.771}{12})} = 473{,}386 \text{ lbm/ft}^2\text{-hr}$$

$$Re = \frac{(0.771/12)(473{,}386)}{1.9983} = 15{,}220$$

Prandtl Number: Pr

$$Pr = \frac{(C_p)(\mu)}{K}$$

From Appendix C,

$$C_p = 1.01881-(0.4802)(10^{-3})(T)+(0.3274)(10^{-5})(T^2)$$

$$-(0.604)(10^{-8})(T^3)$$

At $(t_b)_1 = 83.1^{\circ}F$

$$C_p = 1.01881-(0.4802)(10^{-3})(83.1)+(0.3274)(10^{-5})(83.12)$$

$$-(0.604)(10^{-8})(83.1^3)$$

$$= 0.998 \text{ Btu/lbm-}^{o}\text{F}$$

From Appendix C,

$$K = 0.30289+(0.7029)(10^{-3})(T)-(0.1178)(10^{-5})(T^2)$$

$$-(0.550)(10^{-9})(T^3)$$

At $(t_b)_1 = 83.1^{o}F$

$$K = 0.30289+(0.7029)(10^{-3})(83.1)-(0.1178)(10^{-5})(83.1^2)$$

$$-(0.550)(10^{-9})(83.1^3)$$

$$= 0.353 \text{ Btu/hr-ft-}^{o}\text{F}$$

So, $Pr = \dfrac{(0.998)(1.9983)}{0.353} = 5.65$

Nusselt Number: Nu

$$Nu = \frac{(h)(d_i)}{K}$$

$$= \frac{(520.2)(0.771/12)}{0.353}$$

$$= 94.7$$

Similarily, the other peripheral heat transfer coefficients at station 1 could be calculated in the same manner, and results are presented in Table IX.

TABLE IX

PERIPHERAL HEAT TRANSFER COEFFICIENTS AT STATION 1

	Peripheral Position							
	1	2	3	4	5	6	7	8
h, $Btu/(hr\text{-}ft^2\text{-}^oF$	520.2	557.6	576.1	538.0	507.8	490.3	488.7	494.8

The average heat transfer coefficient at station 1 can be calculated using equation 4.

$$\bar{h} = \frac{1}{8} \sum_{i=1}^{8} (h_i) \tag{4}$$

$$\bar{h} = 1/8(520.2+557.6+576.1+538.0+507.8+490.3+488.7+494.8)$$

$$= 521.7 \ Btu/(hr\text{-}ft^2\text{-}^oF)$$

Comparison of Experimental Data with Literature

1, Sieder-Tate equation;

$$\bar{h} = 0.027 \ (\frac{K}{di})(Re^{0.8})Pr^{0.333})(\frac{\mu}{\mu_w})^{0.14}$$

$$= 0.027(\frac{0.353}{0.771/12})(15,220)^{0.8}(5.65)^{0.333}(\frac{1.9983}{1.5946})^{0.14}$$

$$= 604.4 \ Btu/(hr\text{-}ft^2\text{-}^oF)$$

$$\left\{\begin{array}{l}\text{ratio of heat transfer}\\ \text{coefficients (experimental to}\\ \text{Sieder-Tate)}\end{array}\right\} = \frac{521.7}{604.4} = 0.86$$

2. Eagle-Ferguson Chart

From Appendix C, density of water, ρ:

$$\rho = 0.999986+(0.1890)(10^{-4})(T, {}^{o}C)-(0.5886)(10^{-5})(T, {}^{o}C)^2$$

$$+(0.1548)(10^{-7})(T, {}^{o}C)^3$$

for $(t_b)_1 = 83.1^{o}F = 28.39^{o}C$

$$\rho = 0.999986+(0.1890))10^{-4})(28.39)-(0.5886)(10^{-5})(28.39)^2$$

$$+(0.1548)(10^{-7})(28.39)^3$$

$$= 0.996 \text{ gm/ml}$$

$$= 62.11 \text{ lbm/ft}^3$$

Water velocity at station 1:

$$V = \frac{W}{\rho A}$$

$$V = \frac{1534.8}{(62.11)(\frac{\pi}{4})(\frac{0.771}{12})^2(3600)}$$

$$= 2.12 \text{ ft/sec}$$

From Eagle-Ferguson Chart at V = 2.12 ft/sec and $(t_b)_1 = 83.1^{o}F$:

$$\bar{h} = 527.0 \text{ Btu/(hr-ft}^2\text{-}^{o}\text{F)}$$

$$\left\{\begin{array}{l}\text{ratio of heat transfer} \\ \text{coefficients (experimental} \\ \text{to Eagle-Ferguson)}\end{array}\right\} = \frac{521.7}{527.0} = 0.99$$

APPENDIX E

CALCULATED RESULTS

RUN NUMBER 206

AVERAGE REYNOLDS NUMBER	=	7306.54	
AVERAGE PRANDTL NUMBER	=	5.34	
MASS FLUX	=	215904.25	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	2824.61	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	6162.30	BTU/HR
Q=M*C*(T2-T1)	=	6171.67	BTU/HR
HEAT LOST	=	18.79	BTU/HR
HEAT BALANCE ERROR PERCENT	=	-0.46	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	275.9	254.9	239.5	155.6	114.8	89.6	401.8	288.7	358.4	312.1	296.0
2	295.5	256.8	241.6	268.0	222.0	220.3	374.1	239.6	314.9	293.6	262.1
3	311.0	286.0	284.8	349.7	337.7	344.3	313.0	230.8	265.8	257.7	253.7
4	301.8	306.9	317.6	397.1	399.0	406.8	203.7	211.2	235.8	248.5	252.6
5	288.3	300.4	338.0	403.8	412.3	443.4	82.3	202.7	220.9	236.6	270.4
6	276.5	292.4	293.5	402.9	420.0	455.2	212.1	245.6	261.4	264.3	286.3
7	268.0	270.0	266.7	362.0	356.6	362.8	397.4	262.8	316.9	308.7	313.5
8	263.8	252.7	257.4	260.4	204.5	199.2	457.1	283.8	351.8	320.5	312.1

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	285.1	277.5	279.9	324.9	308.3	315.2	305.2	245.6	290.9	280.3	280.8

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.95	0.91	0.92	1.06	1.01	1.03	0.99	0.80	0.94	0.90	0.90
SIDTAT	0.90	0.86	0.86	1.00	0.94	0.96	0.93	0.74	0.88	0.85	0.84
HEFCHT	1.02	0.98	0.98	1.14	1.08	1.10	1.06	0.85	1.00	0.97	0.96

RUN NUMBER 207

AVERAGE REYNOLDS NUMBER	=	7272.66	
AVERAGE PRANDTL NUMBER	=	5.36	
MASS FLUX	=	215904.25	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	3067.73	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	6649.55	BTU/HR
Q=M*C*(T2-T1)	=	6589.09	BTU/HR
HEAT LOST	=	17.04	BTU/HR
HEAT BALANCE ERROR PERCENT	=	0.66	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	278.9	259.5	248.3	166.1	121.6	95.2	437.5	337.5	387.0	343.9	324.1
2	297.3	266.3	249.4	269.5	228.9	215.7	399.5	275.0	310.3	295.6	275.0
3	321.2	294.8	290.0	365.2	339.8	351.2	320.1	243.9	268.4	258.8	261.4
4	306.3	308.9	331.3	406.3	405.9	419.3	205.3	207.3	241.0	248.2	261.9
5	298.0	316.4	344.2	429.3	434.4	433.8	84.6	207.7	225.0	236.6	275.2
6	285.8	307.0	305.6	413.7	419.9	449.5	212.9	242.4	271.1	268.2	294.8
7	277.5	280.2	284.7	366.6	348.7	363.6	409.8	270.0	314.0	312.8	320.4
8	274.4	268.5	256.6	260.2	210.5	200.7	449.9	294.6	348.6	319.4	316.0

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	292.4	287.7	288.8	334.6	313.7	316.1	315.0	259.8	295.7	285.4	291.1

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.98	0.95	0.95	1.10	1.03	1.03	1.02	0.84	0.96	0.92	0.93
SIDTAT	0.92	0.89	0.89	1.03	0.96	0.96	0.96	0.79	0.90	0.86	0.87
HEFCHT	1.05	1.02	1.02	1.17	1.10	1.10	1.09	0.90	1.02	0.98	0.99

RUN NUMBER 208

```
AVERAGE REYNOLDS NUMBER              = 13395.61
AVERAGE PRANDTL NUMBER               =      6.52
MASS FLUX                            =473385.44  LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX                    =   3683.93  BTU/(SQ.FT-HR)
Q=AMP*VOLT                           =   7941.73  BTU/HR
Q=M*C*(T2-T1)                        =   8243.61  BTU/HR
HEAT LOST                            =    -16.19  BTU/HR
HEAT BALANCE ERROR PERCENT           =     -3.53
```

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	506.8	479.3	462.9	368.3	299.5	254.3	766.4	695.2	624.3	551.7	509.2
2	539.1	474.4	458.9	464.9	418.7	408.3	683.1	603.4	525.5	488.5	437.7
3	540.0	497.0	493.0	573.2	572.3	578.2	538.6	460.4	442.0	425.1	410.5
4	497.4	511.6	540.9	619.0	667.8	714.0	388.7	365.4	373.6	367.9	384.9
5	479.6	512.5	560.0	659.3	701.5	764.0	232.9	323.3	339.2	347.6	393.2
6	458.5	489.5	515.5	659.7	715.0	742.7	404.9	432.0	417.7	414.8	441.4
7	467.4	469.9	471.6	581.4	583.1	636.2	629.3	528.9	498.1	499.1	492.4
8	470.2	455.1	461.2	464.1	401.2	399.6	736.0	642.7	577.5	540.1	502.4

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	494.9	486.1	495.5	548.7	544.9	562.2	547.5	506.4	474.8	454.4	446.5

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.95	0.93	0.94	1.04	1.03	1.06	1.03	0.95	0.89	0.85	0.83
SIDTAT	0.91	0.88	0.90	0.99	0.98	1.01	0.98	0.91	0.85	0.81	0.79
HEFCHT	1.01	0.98	1.00	1.11	1.10	1.13	1.10	1.01	0.95	0.91	0.89

RUN NUMBER 209

AVERAGE REYNOLDS NUMBER	= 16576.16	
AVERAGE PRANDTL NUMBER	= 5.14	
MASS FLUX	=473385.44	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	= 10404.93	BTU/(SQ.FT-HR)
Q=AMP*VOLT	= 22773.91	BTU/HR
Q=M*C*(T2-T1)	= 23726.84	BTU/HR
HEAT LOST	= 20.65	BTU/HR
HEAT BALANCE ERROR PERCENT	= -4.19	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	521.1	489.2	463.8	383.6	329.2	294.2	814.3	775.4	724.4	690.2	625.2
2	557.6	491.7	459.9	481.4	437.5	432.6	715.1	659.3	629.2	600.5	547.5
3	576.1	520.3	504.4	574.7	553.6	601.0	577.7	524.3	515.5	513.8	500.1
4	538.0	535.6	534.8	618.7	646.7	705.4	425.8	412.7	436.2	433.5	463.5
5	507.8	514.3	566.6	632.7	658.6	750.5	280.9	361.5	384.2	401.2	454.7
6	490.3	519.9	530.1	641.1	692.5	740.5	466.4	489.2	485.5	489.3	531.3
7	488.7	483.6	486.4	588.6	573.8	616.9	694.8	619.9	604.5	627.0	597.7
8	494.8	486.9	469.6	480.3	429.4	425.1	825.9	752.4	708.9	665.4	615.1

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	521.8	505.2	502.0	550.1	540.2	570.8	600.1	574.3	561.1	552.6	541.9

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.93	0.88	0.87	0.95	0.93	0.97	1.02	0.97	0.94	0.93	0.89
SIDTAT	0.86	0.81	0.80	0.88	0.85	0.90	0.94	0.89	0.87	0.85	0.82
HEFCHT	0.99	0.94	0.93	1.01	0.99	1.04	1.08	1.03	1.00	0.99	0.95

RUN NUMBER 210

AVERAGE REYNOLDS NUMBER	=	17064.03	
AVERAGE PRANDTL NUMBER	=	6.06	
MASS FLUX	=	565083.06	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	3385.06	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	7296.84	BTU/HR
Q=M*C*(T2-T1)	=	7641.82	BTU/HR
HEAT LOST	=	-4.73	BTU/HR
HEAT BALANCE ERROR PERCENT	=	-4.55	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	596.2	538.2	514.4	415.5	357.4	322.1	871.4	798.1	784.4	713.2	656.1
2	615.6	535.2	500.9	521.3	470.5	463.9	775.7	681.3	652.3	643.4	582.8
3	630.5	555.8	537.7	606.3	583.2	642.2	608.7	547.6	548.2	547.9	528.3
4	596.2	565.0	576.5	648.0	664.5	727.6	463.6	432.3	445.1	458.5	476.9
5	532.5	566.2	600.0	668.7	689.5	800.6	299.5	384.9	402.4	419.7	483.9
6	517.1	546.8	565.7	661.3	714.2	768.0	497.1	506.8	505.5	521.6	553.1
7	527.2	516.8	527.5	631.0	605.0	641.1	754.9	644.1	632.9	648.2	659.9
8	542.1	528.7	511.5	529.0	468.7	456.4	874.7	758.1	724.3	694.5	665.2

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	569.6	544.1	541.8	585.1	569.1	602.7	643.2	594.2	586.9	580.9	575.3

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.92	0.87	0.87	0.94	0.91	0.96	1.02	0.94	0.93	0.92	0.91
SIDTAT	0.88	0.83	0.82	0.89	0.86	0.91	0.97	0.90	0.89	0.87	0.86
HEFCHT	0.98	0.93	0.92	0.99	0.96	1.02	1.09	1.00	0.99	0.98	0.97

RUN NUMBER 211

AVERAGE REYNOLDS NUMBER	=	19844.54	
AVERAGE PRANDTL NUMBER	=	5.12	
MASS FLUX	=	565083.06	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	14248.03	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	30919.33	BTU/HR
Q=M*C*(T2-T1)	=	32139.99	BTU/HR
HEAT LOST	=	19.03	BTU/HR
HEAT BALANCE ERROR PERCENT	=	-3.93	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	650.7	626.3	594.4	496.5	434.7	391.2	1078.6	1006.1	915.5	836.7	740.3
2	682.2	618.8	593.0	612.4	560.1	550.2	923.9	841.1	764.3	720.2	659.3
3	890.0	638.3	634.1	734.1	725.3	761.6	728.2	655.8	631.6	614.5	533.8
4	629.0	646.4	674.0	800.3	854.9	929.6	548.7	516.1	518.7	515.8	500.2
5	598.5	654.7	710.8	835.7	886.1	1002.2	365.8	447.1	458.3	469.4	619.7
6	574.8	632.6	650.9	832.1	904.2	993.9	595.4	620.4	587.1	593.2	734.1
7	587.2	602.7	607.4	760.2	759.3	813.1	897.9	784.7	744.4	754.9	720.0
8	599.7	596.8	586.5	604.8	553.1	550.1	1063.0	955.2	876.9	822.1	757.9

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT,HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	651.5	627.1	631.4	709.5	709.7	749.0	775.2	728.3	687.1	665.9	658.1

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	1.01	0.95	0.95	1.06	1.06	1.11	1.14	1.07	1.00	0.96	0.94
SIDTAT	0.94	0.87	0.87	0.98	0.97	1.02	1.05	0.98	0.92	0.88	0.85
HEFCHT	1.08	1.01	1.01	1.13	1.12	1.18	1.21	1.13	1.06	1.03	0.99

RUN NUMBER 212

AVERAGE REYNOLDS NUMBER	=	19352.41	
AVERAGE PRANDTL NUMBER	=	6.23	
MASS FLUX	=	656780.69	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	4305.91	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	9403.82	BTU/HR
Q=M*C*(T2-T1)	=	9946.20	BTU/HR
HEAT LOST	=	-9.59	BTU/HR
HEAT BALANCE ERROR PERCENT	=	-5.50	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	636.1	597.5	566.9	479.0	415.4	378.7	1069.0	1002.8	971.4	892.4	794.0
2	678.2	603.2	581.3	607.1	562.7	550.2	909.2	861.8	797.0	775.3	683.6
3	719.2	646.0	627.3	713.8	715.6	757.7	728.3	684.0	675.7	657.5	616.0
4	656.4	642.2	667.1	768.7	822.0	909.1	555.7	521.0	547.1	541.7	554.1
5	608.2	643.0	692.5	783.5	846.0	985.2	359.0	462.0	476.4	484.4	551.7
6	581.7	634.5	645.2	801.5	880.8	978.4	602.5	624.7	606.5	627.8	643.6
7	592.3	605.0	609.6	739.3	742.2	802.9	905.7	796.1	774.5	812.9	766.1
8	609.0	586.7	564.4	615.4	552.8	547.9	1070.8	1009.3	924.2	889.5	786.7

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	635.1	619.8	619.3	688.5	692.2	738.8	775.0	745.2	721.6	710.2	674.5

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.92	0.89	0.89	0.99	0.99	1.06	1.11	1.06	1.03	1.01	0.95
SIDTAT	0.88	0.85	0.85	0.94	0.94	1.00	1.05	1.01	0.98	0.96	0.91
HEFCHT	0.98	0.95	0.94	1.05	1.05	1.12	1.17	1.13	1.09	1.07	1.01

RUN NUMBER 213

AVERAGE REYNOLDS NUMBER	=	22369.27	
AVERAGE PRANDTL NUMBER	=	5.30	
MASS FLUX	=	656780.69	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	15085.61	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	33002.11	BTU/HR
Q=M*C*(T2-T1)	=	34614.22	BTU/HR
HEAT LOST	=	18.66	BTU/HR
HEAT BALANCE ERROR PERCENT	=	-4.82	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	710.3	682.4	633.0	546.9	485.5	439.6	1164.7	1087.3	1002.2	927.7	819.7
2	753.8	678.2	645.8	672.2	620.8	608.9	1001.7	924.6	849.4	799.3	719.3
3	778.3	708.3	694.6	794.4	779.3	824.4	802.4	721.6	695.1	674.0	649.3
4	709.4	710.3	731.9	851.9	911.9	1000.6	610.9	568.3	567.8	558.3	578.4
5	651.4	703.7	758.9	897.8	945.0	1081.3	410.0	493.8	496.8	506.0	556.6
6	621.9	687.7	703.0	886.4	979.4	1071.5	666.0	676.4	638.4	644.1	665.6
7	645.8	660.3	657.9	817.6	820.9	876.3	989.6	871.9	825.8	828.7	815.6
8	656.3	660.9	638.9	667.4	612.8	603.7	1177.5	1038.9	973.8	918.7	839.8

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	690.9	686.5	683.0	766.8	769.5	813.3	852.9	797.9	756.2	732.1	705.5

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.96	0.93	0.92	1.03	1.03	1.08	1.13	1.05	0.99	0.95	0.91
SIDTAT	0.89	0.86	0.85	0.95	0.95	1.00	1.04	0.97	0.91	0.88	0.83
HEFCHT	1.02	0.99	0.98	1.10	1.09	1.15	1.20	1.12	1.05	1.02	0.96

RUN NUMBER 214

AVERAGE REYNOLDS NUMBER	=	23942.32	
AVERAGE PRANDTL NUMBER	=	5.68	
MASS FLUX	=	748478.31	LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX	=	12914.80	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	28191.01	BTU/HR
Q=M*C*(T2-T1)	=	28879.81	BTU/HR
HEAT LOST	=	7.95	BTU/HR
HEAT BALANCE ERROR PERCENT	=	-2.44	

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	730.2	701.4	654.8	574.3	500.3	463.6	1239.4	1113.4	1023.9	951.0	853.1
2	792.1	701.6	675.5	738.3	685.7	658.2	1073.7	954.2	885.6	856.8	773.5
3	852.3	769.7	759.5	870.1	843.6	888.7	849.1	759.9	742.9	730.3	711.8
4	783.3	774.5	798.5	936.5	992.7	1075.7	648.0	608.3	609.6	601.8	631.7
5	704.1	780.8	841.2	972.1	1022.3	1172.9	441.8	532.0	531.5	545.3	602.7
6	686.3	761.3	783.6	975.9	1067.1	1150.3	706.6	725.9	686.6	696.7	730.9
7	705.9	734.5	734.9	909.5	888.2	937.2	1026.3	913.9	885.6	906.8	891.3
8	702.0	708.8	681.4	740.8	671.4	652.2	1203.6	1083.7	1014.8	982.4	902.4

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	744.5	741.6	741.2	839.7	833.9	874.8	898.6	836.4	797.6	783.9	762.2

RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.95	0.93	0.93	1.05	1.03	1.08	1.11	1.03	0.97	0.95	0.92
SIDTAT	0.89	0.87	0.86	0.98	0.96	1.01	1.03	0.95	0.91	0.89	0.85
HEFCHT	1.01	0.99	0.98	1.11	1.10	1.15	1.17	1.09	1.04	1.01	0.97

RUN NUMBER 215

```
AVERAGE REYNOLDS NUMBER          = 27321.78
AVERAGE PRANDTL NUMBER           =     5.58
MASS FLUX                        =840175.94  LBM/(SQ.FT-HR)
AVERAGE HEAT FLUX                = 16584.82  BTU/(SQ.FT-HR)
Q=AMP*VOLT                       = 36373.30  BTU/HR
Q=M*C*(T2-T1)                    = 37559.80  BTU/HR
HEAT LOST                        =     1.61  BTU/HR
HEAT BALANCE ERROR PERCENT       =    -3.21
```

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
1	781.7	752.6	710.8	630.3	553.2	507.1	1426.0	1305.3	1183.8	1070.1	958.0
2	848.4	760.6	736.2	832.8	760.6	731.0	1215.5	1110.0	1004.5	952.7	865.3
3	924.6	841.1	829.4	962.5	933.1	982.0	963.6	873.5	839.1	814.2	795.0
4	841.5	830.5	881.1	1027.3	1102.8	1236.8	733.2	683.4	670.5	658.1	684.0
5	747.0	827.0	902.1	1064.2	1122.6	1301.2	498.3	589.5	575.8	583.9	638.6
6	720.5	814.8	856.9	1080.9	1181.7	1333.9	801.5	817.3	756.3	757.7	795.0
7	754.2	795.2	803.3	989.1	993.7	1062.1	1204.8	1047.2	1000.4	1029.6	1025.3
8	744.8	767.8	734.1	869.4	753.9	738.6	1427.9	1264.1	1165.2	1116.2	1025.4

AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.F)

	1	2	3	4	5	6	7	8	9	10	11
HAVG	795.3	798.7	806.8	932.1	925.2	986.6	1033.8	961.3	899.4	872.8	848.3

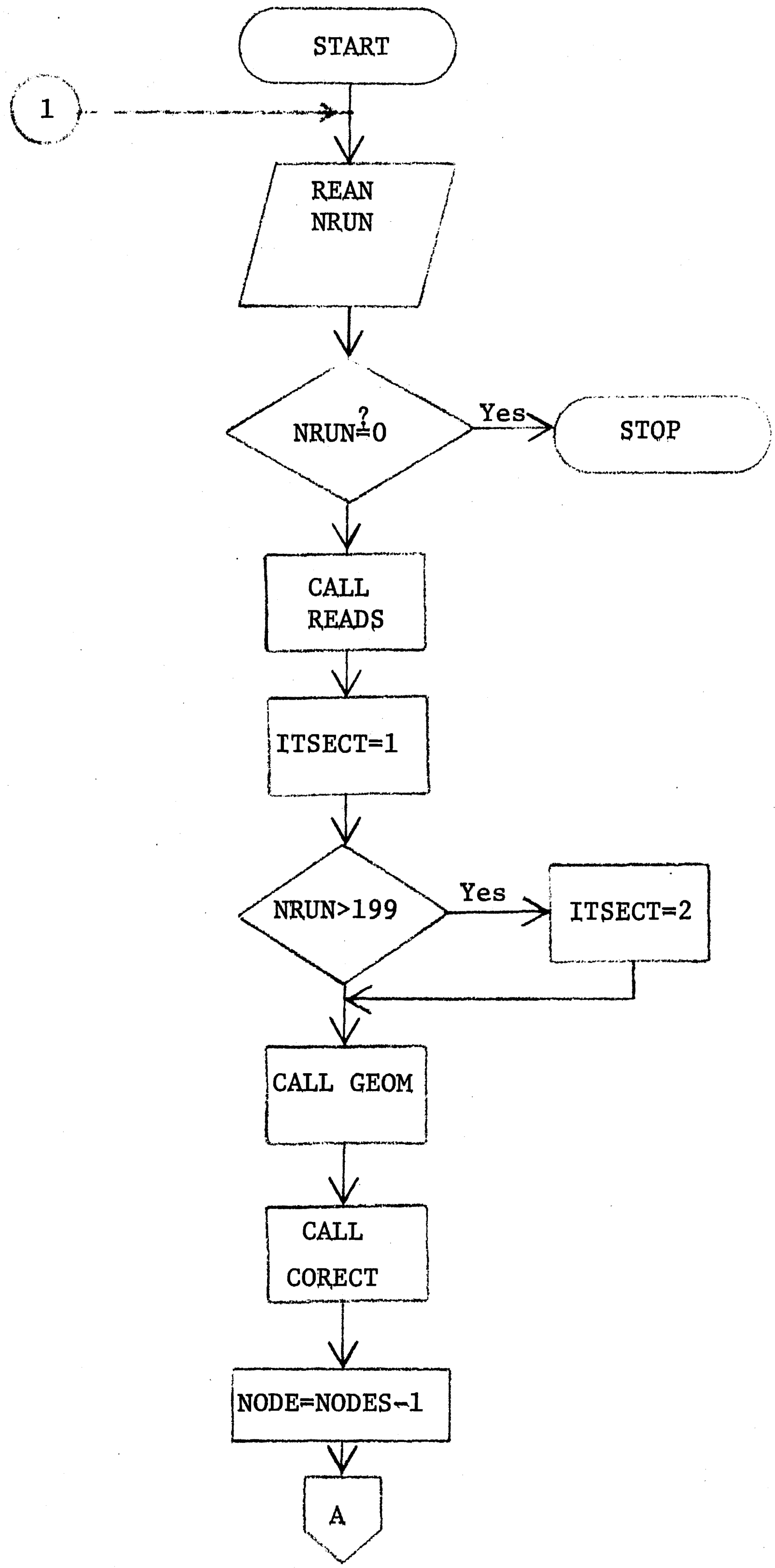
RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOSE PREDICTED BY LITERATURE

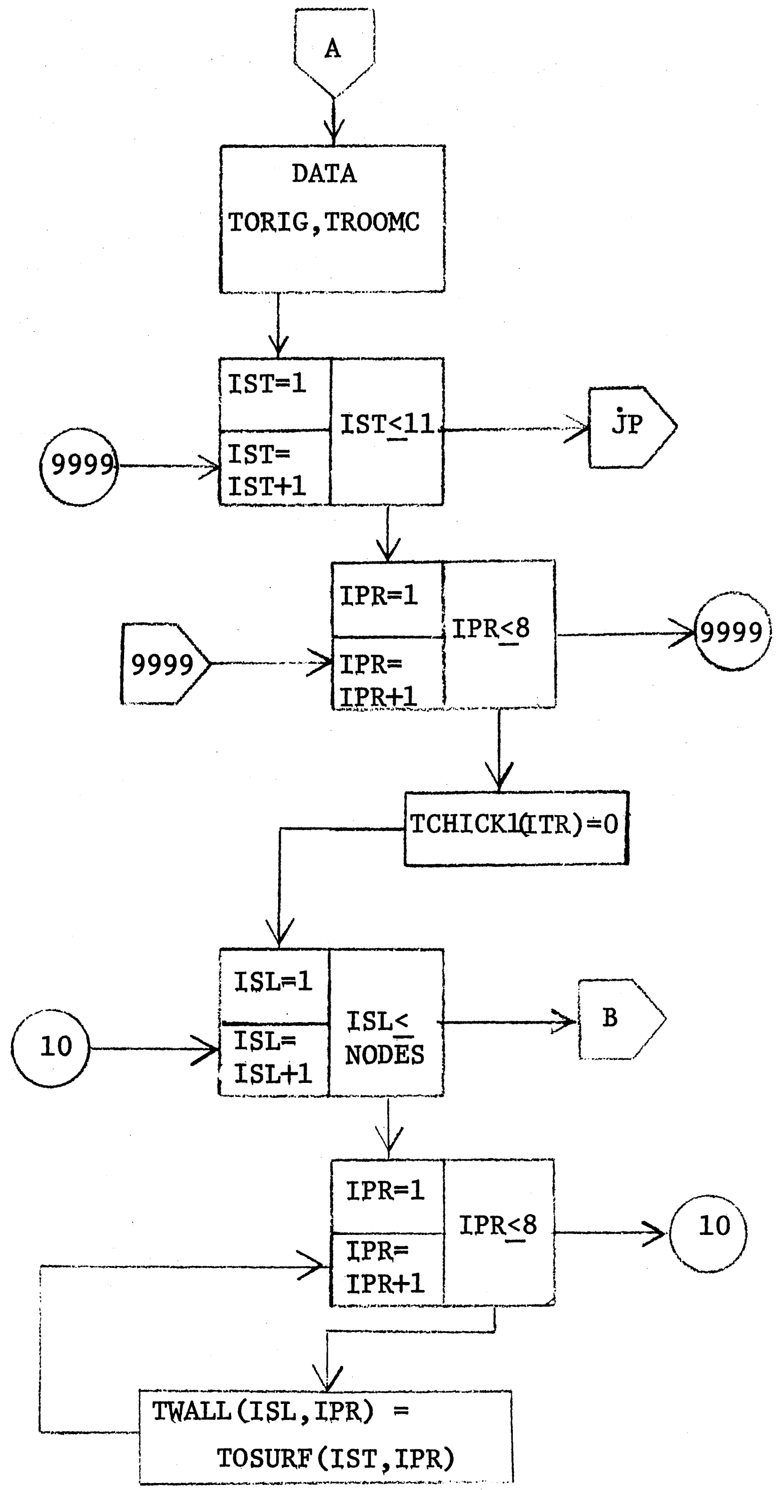
	1	2	3	4	5	6	7	8	9	10	11
DITBL	0.93	0.91	0.91	1.05	1.04	1.10	1.15	1.06	0.99	0.96	0.92
SIDTAT	0.86	0.84	0.85	0.98	0.96	1.02	1.07	0.99	0.92	0.89	0.85
HEFCHT	0.98	0.96	0.97	1.12	1.10	1.17	1.22	1.13	1.05	1.02	0.98

APPENDIX F

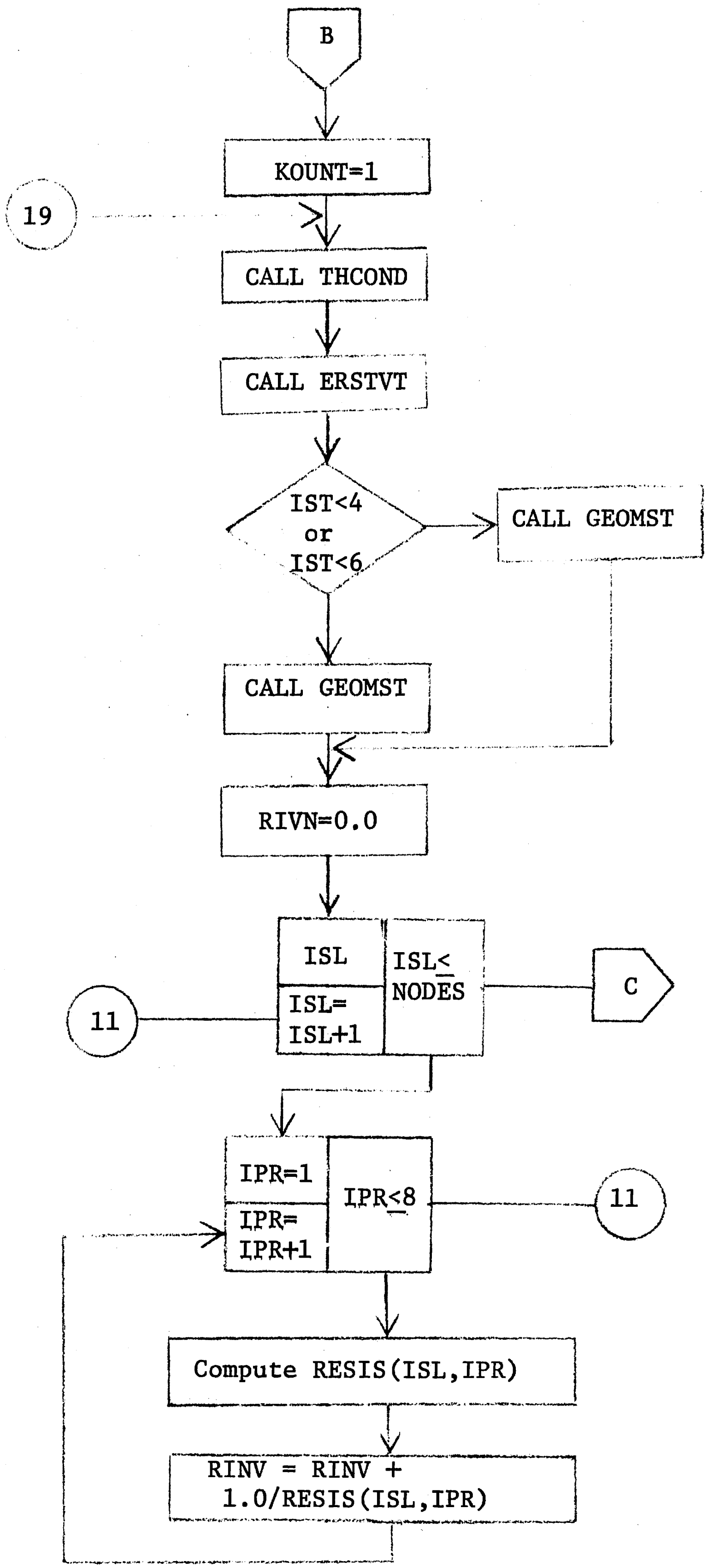
COMPUTER PROGRAM LISTING

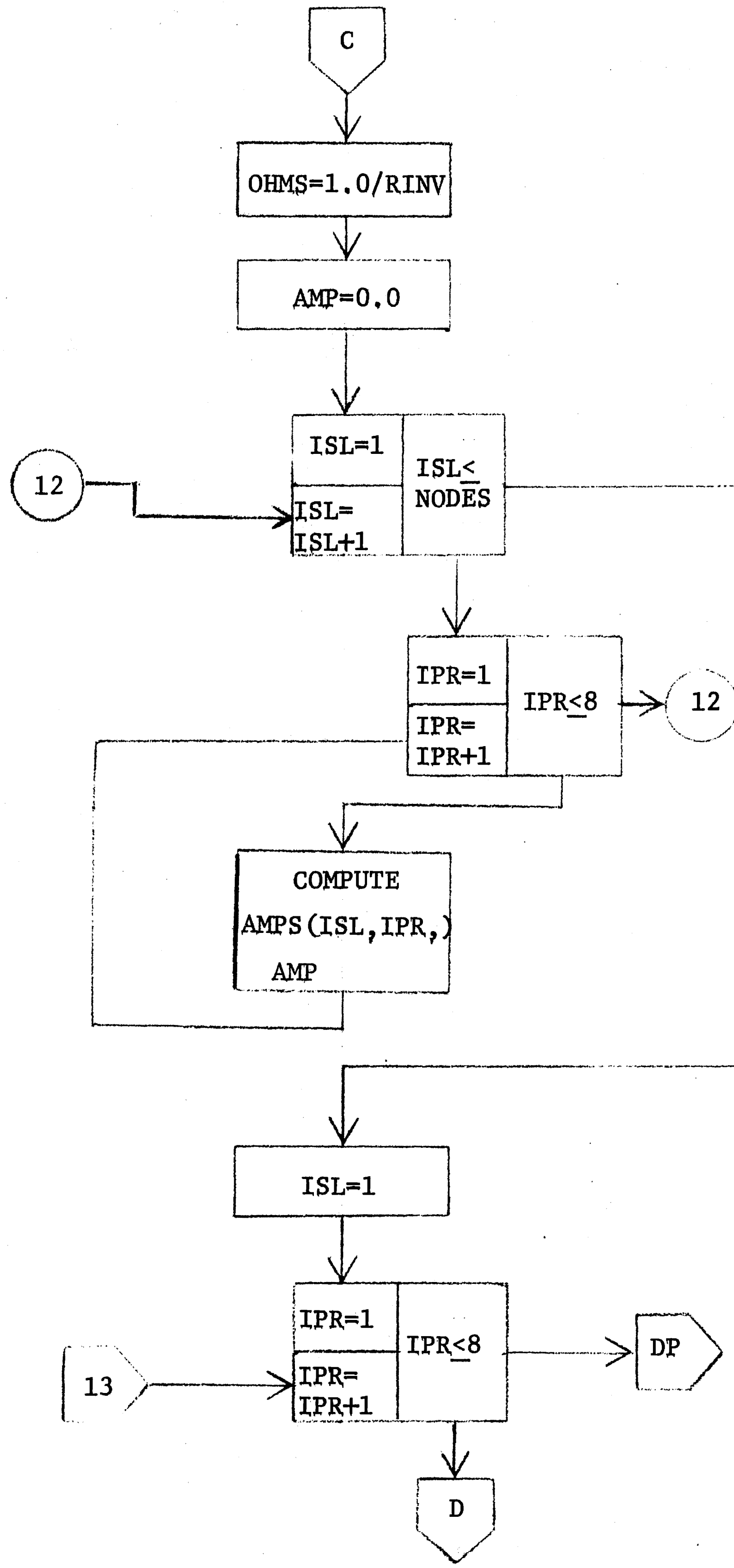
AND FLOW CHARTS

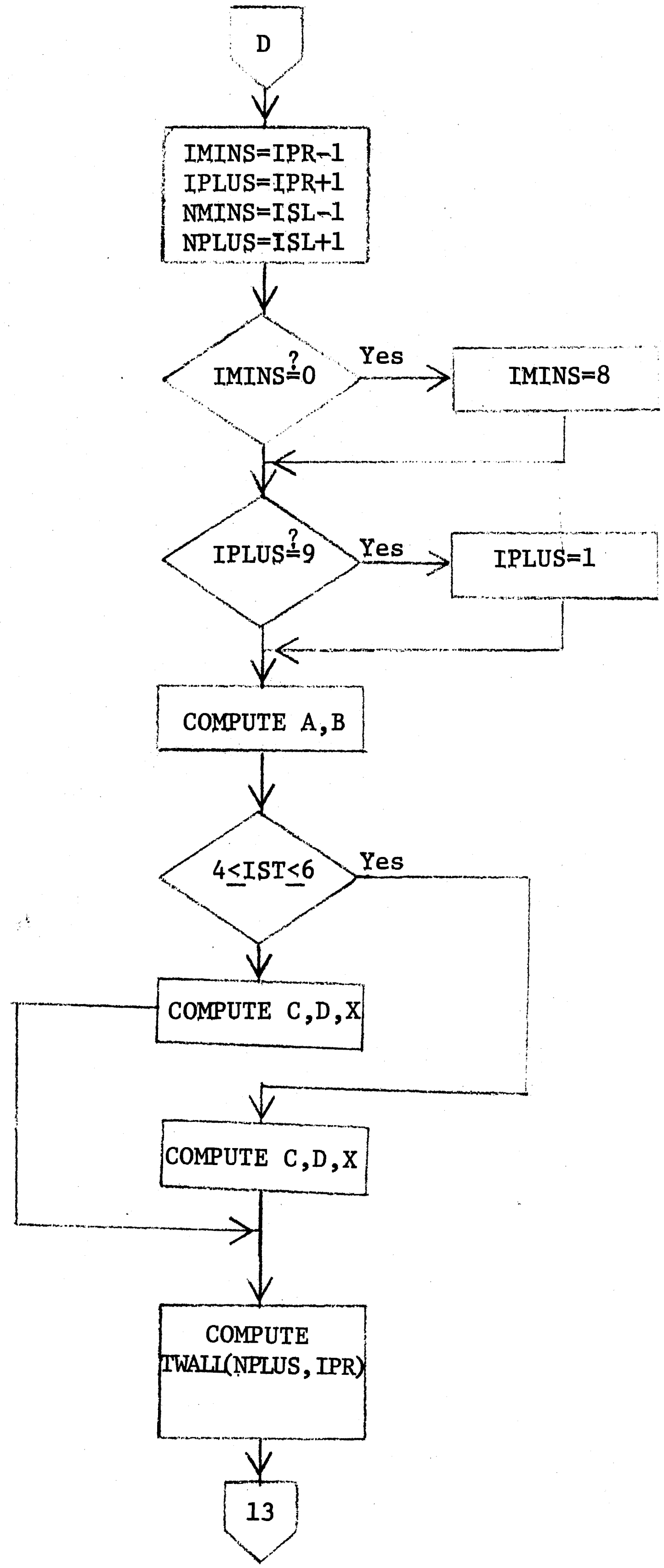

START
1
REAN
NRUN
NRUN?=0
Yes
STOP
CALL
READS
ITSECT=1
NRUN>199
Yes
ITSECT=2
CALL GEOM
CALL
CORECT
NODE=NODES-1
A

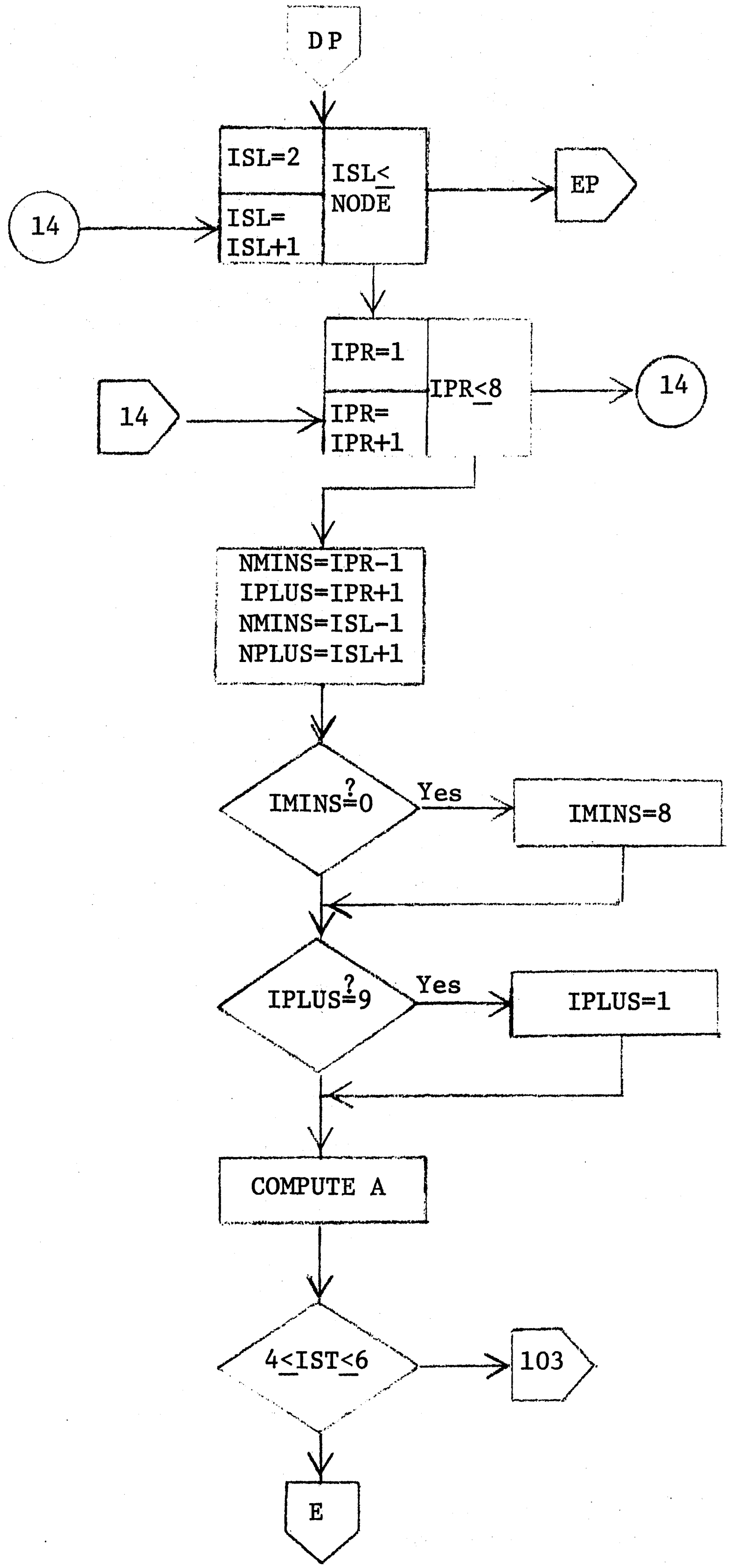


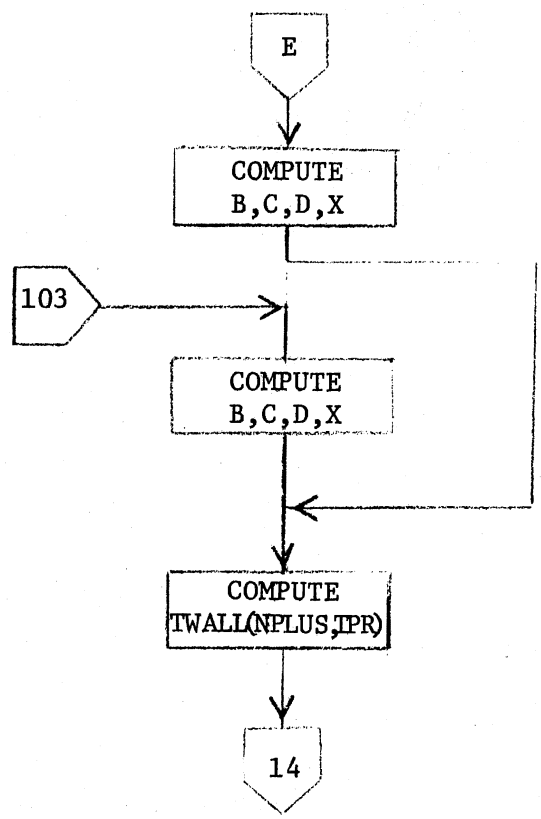
A
DATA
TORIG,TROOMC
IST=1
IST=
IST+1
IST≤11
JP
9999
IPR=1
IPR=
IPR+1
IPR≤8
9999
9999
TCHICK1(ITR)=0
ISL=1
ISL=
ISL+1
ISL≤
NODES
10
B
IPR=1
IPR=
IPR+1
IPR≤8
10
TWALL(ISL,IPR) =
TOSURF(IST,IPR)

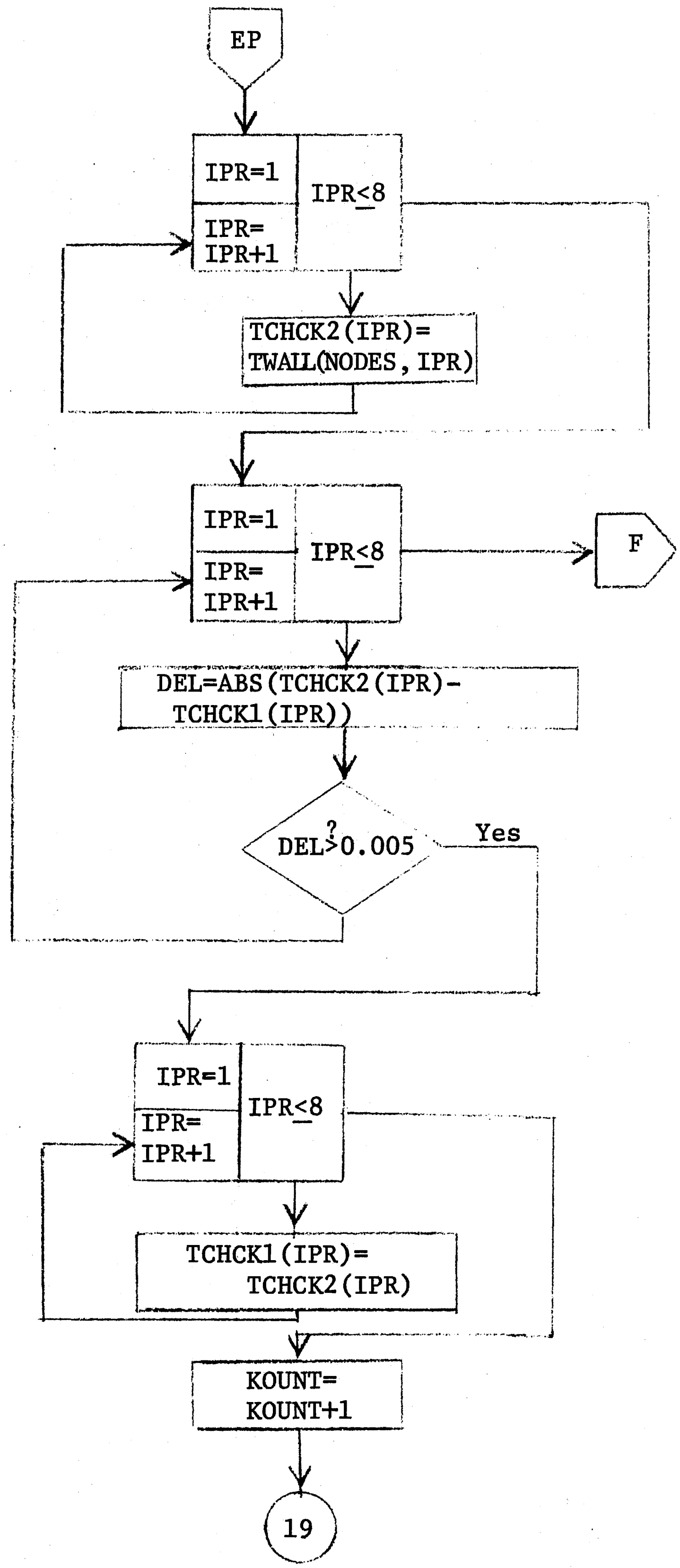

B
KOUNT=1
19
CALL THCOND
CALL ERSTVT
IST<4
or
IST<6
CALL GEOMST
CALL GEOMST
RIVN=0.0
ISL
ISL=
ISL+1
ISL≤
NODES
C
11
IPR=1
IPR=
IPR+1
IPR≤8
11
Compute RESIS(ISL,IPR)
RINV = RINV +
1.0/RESIS(ISL,IPR)

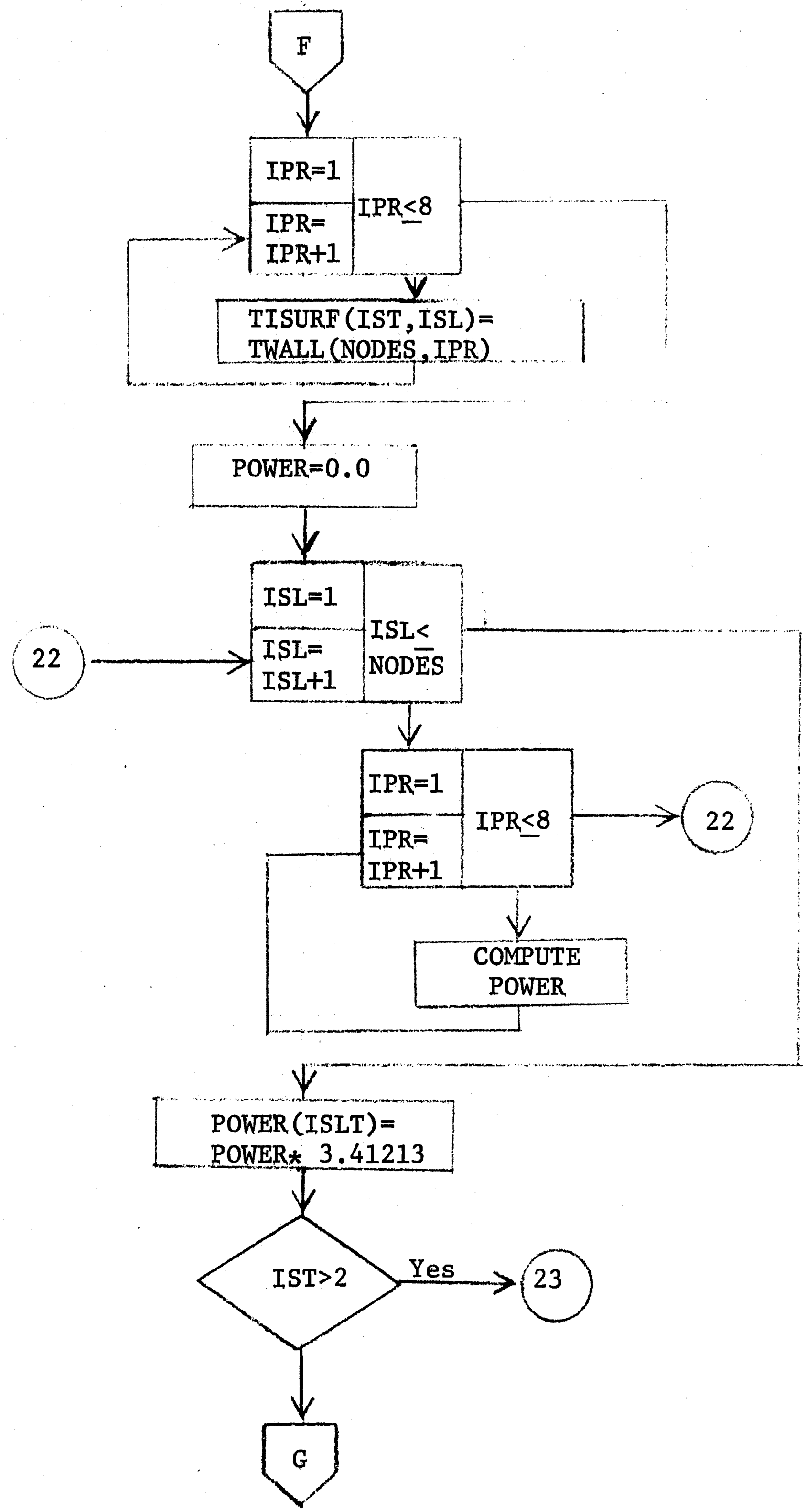

C
OHMS=1.0/RINV
AMP=0.0
ISL=1
ISL=
ISL+1
ISL≤
NODES
12
IPR=1
IPR=
IPR+1
IPR≤8
12
COMPUTE
AMPS(ISL,IPR,)
AMP
ISL=1
IPR=1
IPR=
IPR+1
IPR≤8
13
DP
D


D
IMINS=IPR-1
IPLUS=IPR+1
NMINS=ISL-1
NPLUS=ISL+1
IMINS≟0
Yes
IMINS=8
IPLUS≟9
Yes
IPLUS=1
COMPUTE A,B
4≤IST≤6
Yes
COMPUTE C,D,X
COMPUTE C,D,X
COMPUTE
TWALL(NPLUS,IPR)
13

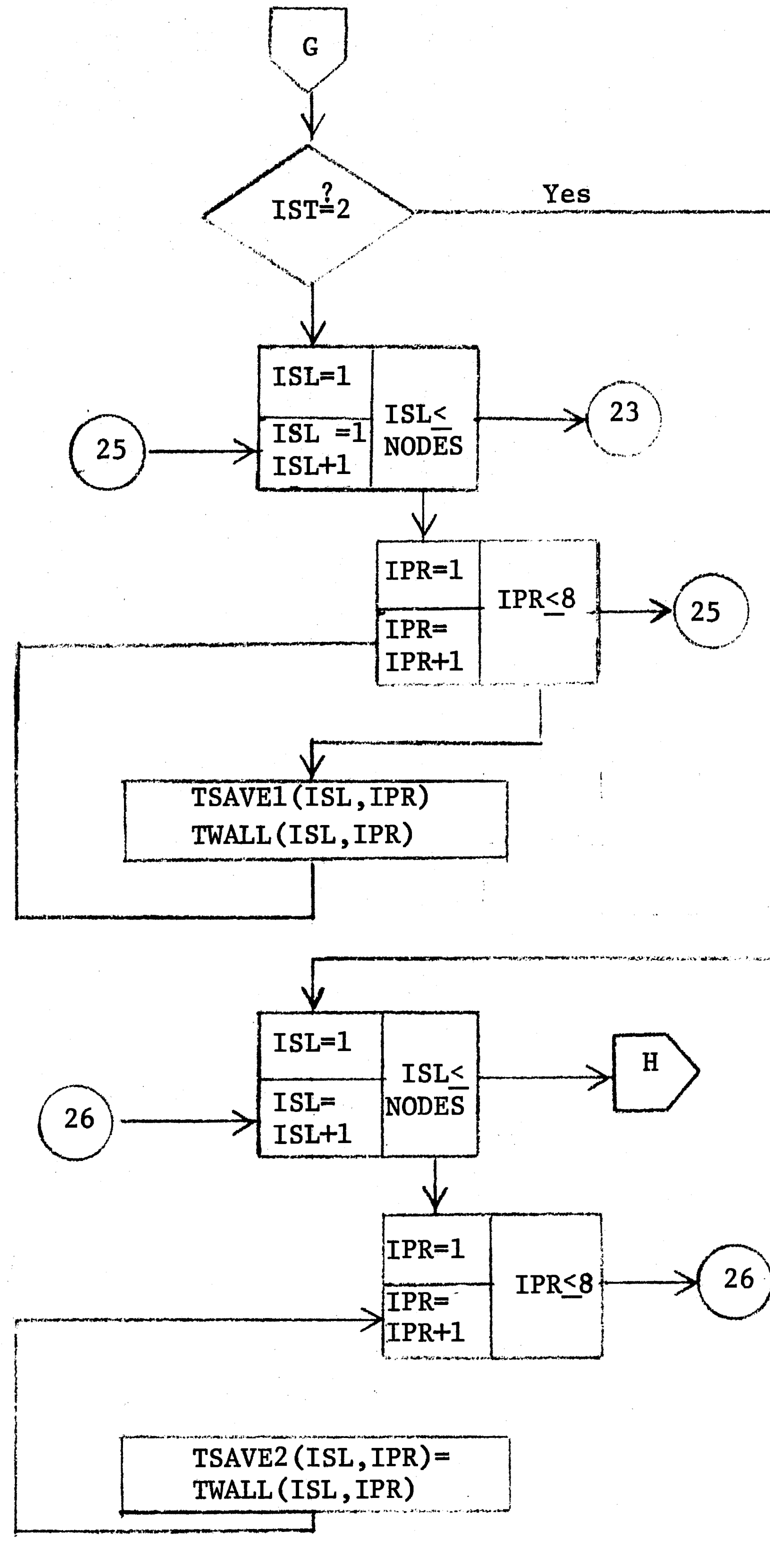

DP
ISL=2
ISL=
ISL+1
ISL<
NODE
EP
14
IPR=1
IPR=
IPR+1
IPR<8
14
14
NMINS=IPR-1
IPLUS=IPR+1
NMINS=ISL-1
NPLUS=ISL+1
IMINS?=0
Yes
IMINS=8
IPLUS?=9
Yes
IPLUS=1
COMPUTE A
4<IST<6
103
E

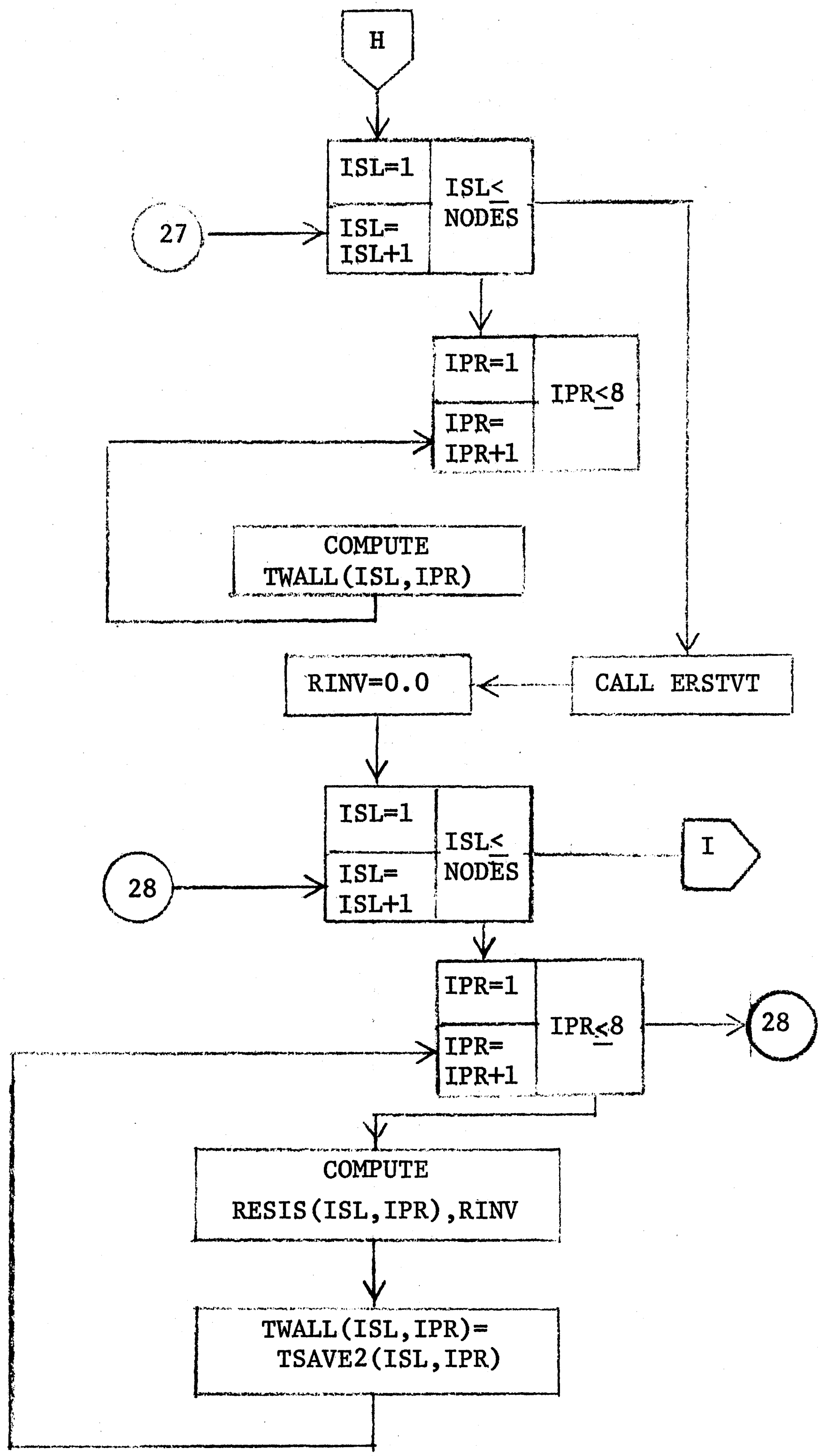

E
COMPUTE
B,C,D,X
103
COMPUTE
B,C,D,X
COMPUTE
TWALL(NPLUS,TPR)
14

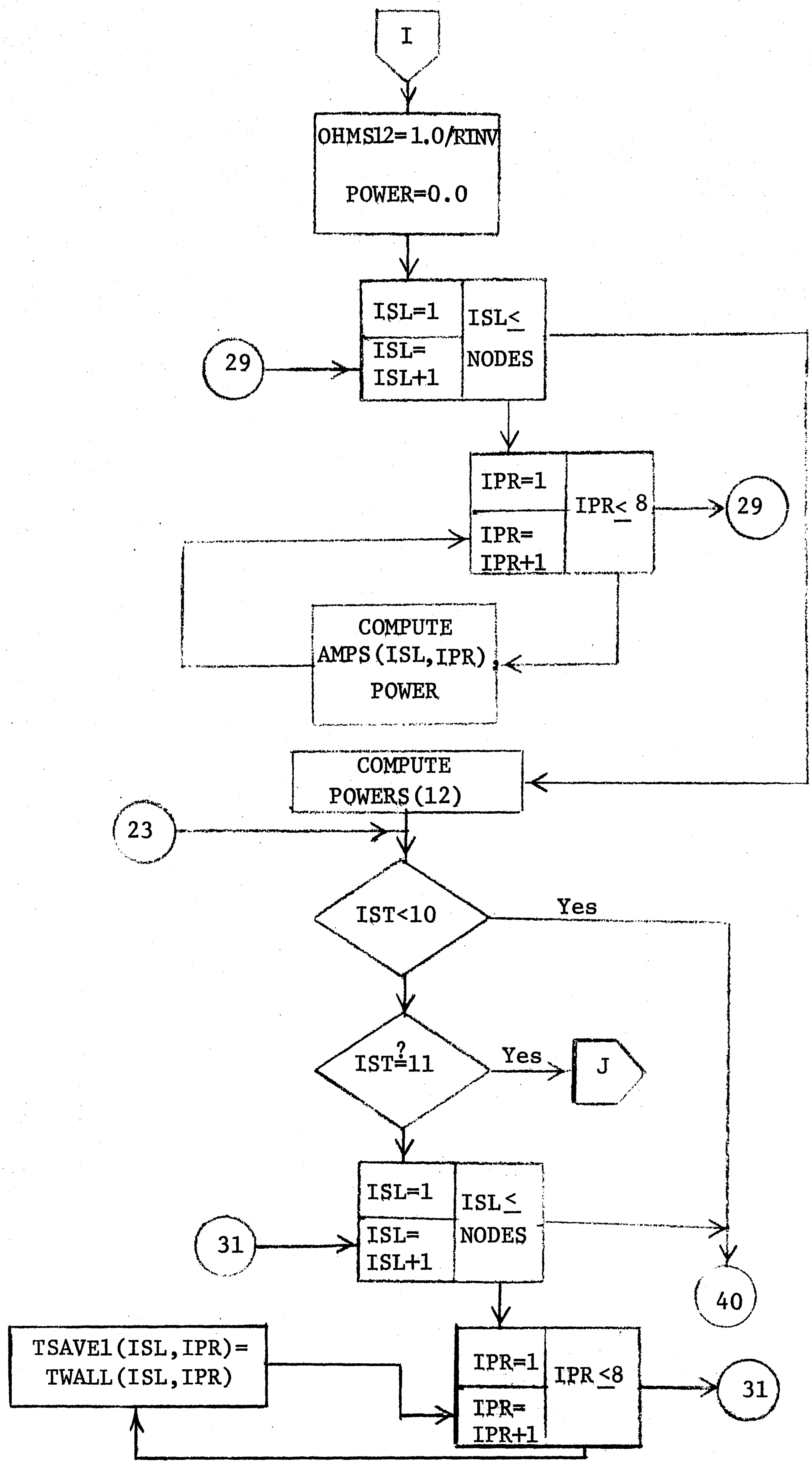

EP
IPR=1
IPR=
IPR+1
IPR<8
TCHCK2(IPR)=
TWALL(NODES,IPR)
IPR=1
IPR=
IPR+1
IPR<8
F
DEL=ABS(TCHCK2(IPR)-
TCHCK1(IPR))
DEL?>0.005
Yes
IPR=1
IPR=
IPR+1
IPR<8
TCHCK1(IPR)=
TCHCK2(IPR)
KOUNT=
KOUNT+1
19

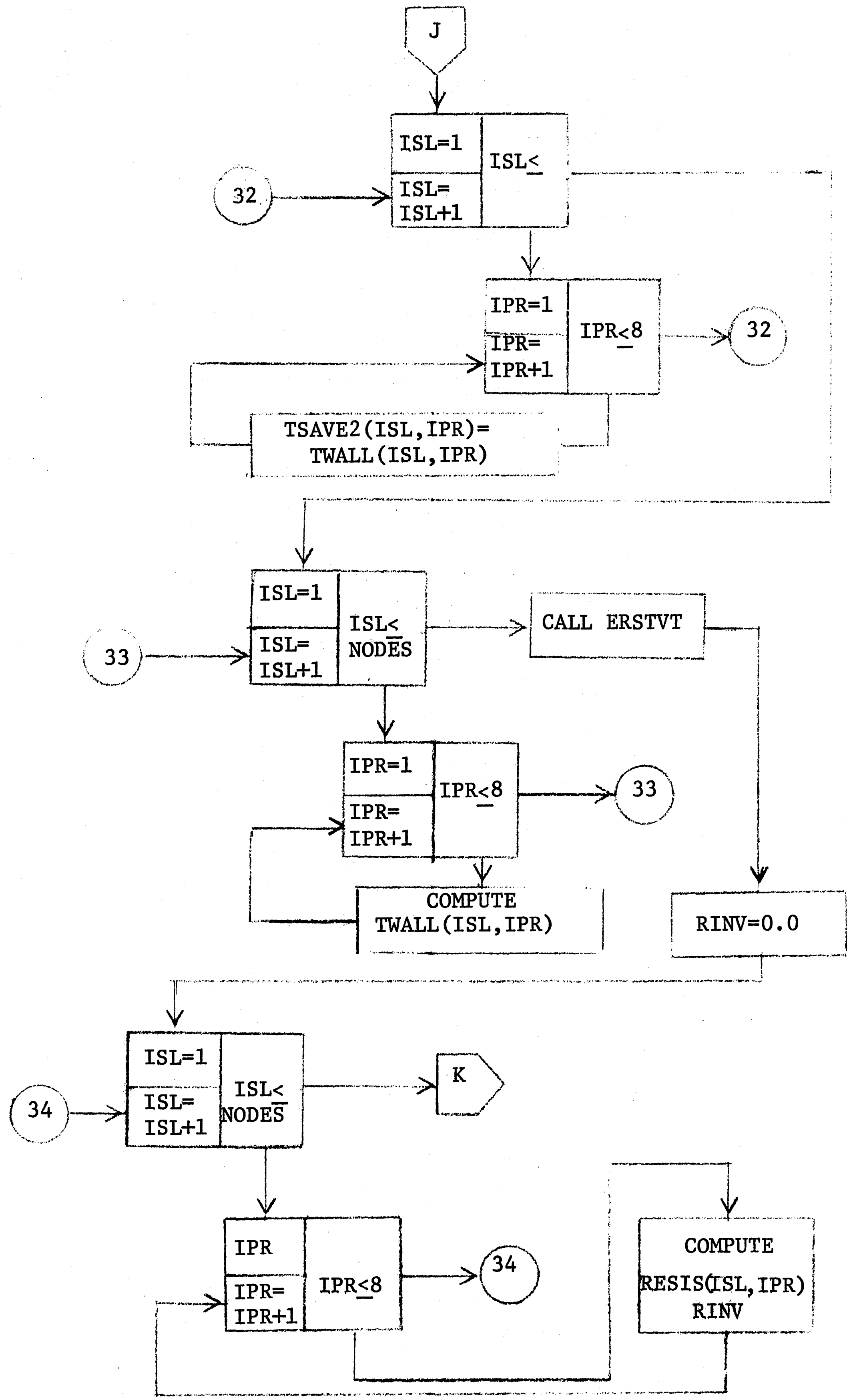


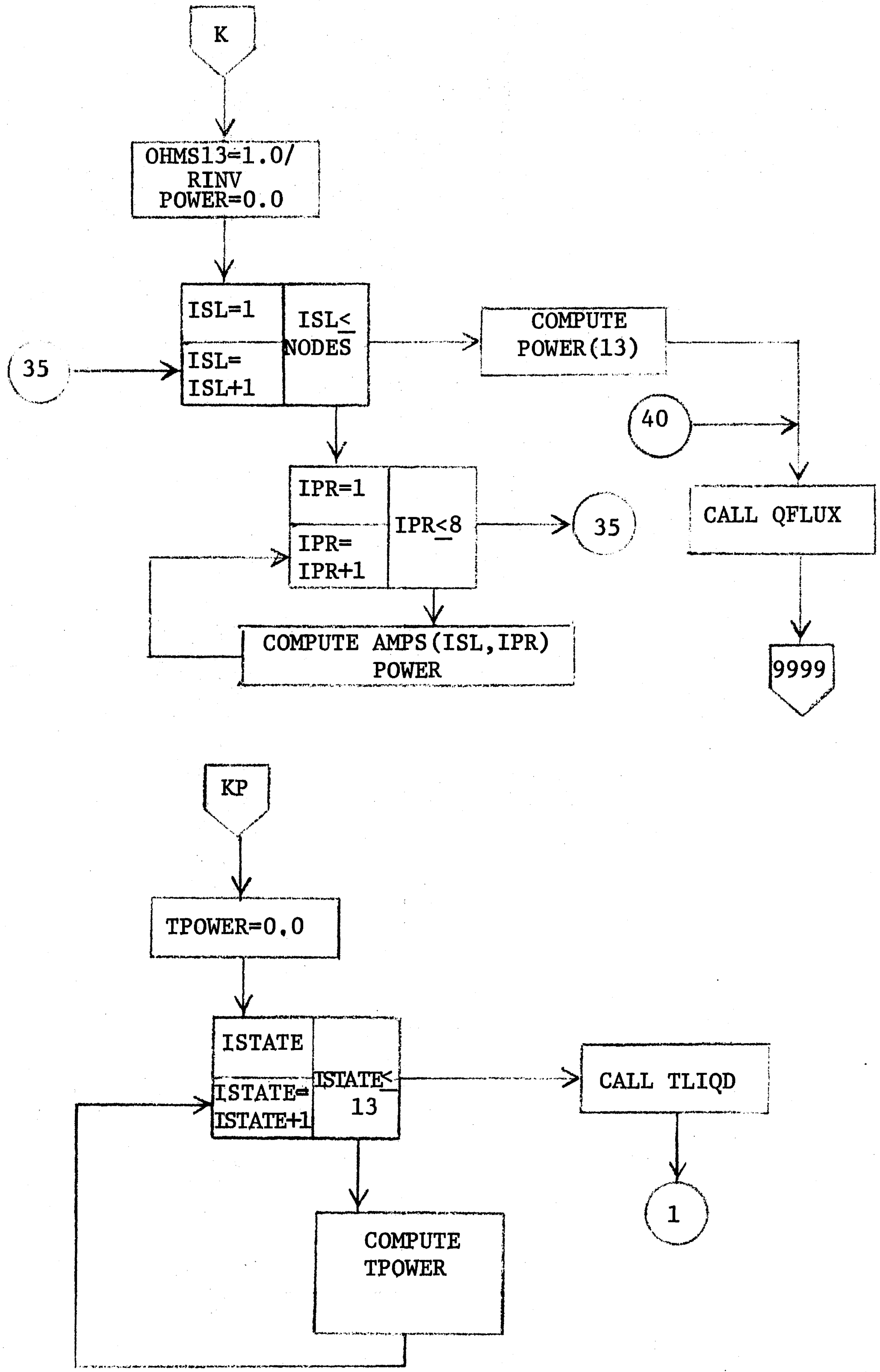
F
IPR=1
IPR=
IPR+1
IPR≤8
TISURF(IST,ISL)=
TWALL(NODES,IPR)
POWER=0.0
ISL=1
22
ISL=
ISL+1
ISL≤
NODES
IPR=1
IPR=
IPR+1
IPR≤8
22
COMPUTE
POWER
POWER(ISLT)=
POWER* 3.41213
IST>2
Yes
23
G


G
IST=2 ?
Yes
ISL=1
ISL =1 ISL+1
ISL≤ NODES
25
23
IPR=1
IPR= IPR+1
IPR≤8
25
TSAVE1(ISL,IPR)
TWALL(ISL,IPR)
ISL=1
ISL= ISL+1
ISL≤ NODES
26
H
IPR=1
IPR= IPR+1
IPR≤8
26
TSAVE2(ISL,IPR)=
TWALL(ISL,IPR)


H
ISL=1
ISL=
ISL+1
ISL≤
NODES
27
IPR=1
IPR=
IPR+1
IPR≤8
COMPUTE
TWALL(ISL,IPR)
RINV=0.0
CALL ERSTVT
ISL=1
ISL=
ISL+1
ISL≤
NODES
28
I
IPR=1
IPR=
IPR+1
IPR≤8
28
COMPUTE
RESIS(ISL,IPR),RINV
TWALL(ISL,IPR)=
TSAVE2(ISL,IPR)


I
OHMS12=1.0/RINV
POWER=0.0
ISL=1
ISL=
ISL+1
ISL≤
NODES
29
IPR=1
IPR=
IPR+1
IPR≤ 8
29
COMPUTE
AMPS(ISL,IPR),
POWER
COMPUTE
POWERS(12)
23
IST<10
Yes
IST=?11
Yes
J
ISL=1
ISL=
ISL+1
ISL≤
NODES
31
40
TSAVE1(ISL,IPR)=
TWALL(ISL,IPR)
IPR=1
IPR=
IPR+1
IPR≤8
31


J
ISL=1
ISL=
ISL+1
ISL≤
32
IPR=1
IPR=
IPR+1
IPR≤8
32
TSAVE2(ISL,IPR)=
TWALL(ISL,IPR)
ISL=1
ISL=
ISL+1
ISL≤
NODES
33
CALL ERSTVT
IPR=1
IPR=
IPR+1
IPR≤8
33
COMPUTE
TWALL(ISL,IPR)
RINV=0.0
ISL=1
ISL=
ISL+1
ISL≤
NODES
34
K
IPR
IPR=
IPR+1
IPR≤8
34
COMPUTE
RESIS(ISL,IPR)
RINV


K
OHMS13=1.0/ RINV
POWER=0.0
ISL=1
ISL= ISL+1
ISL≤ NODES
35
COMPUTE POWER(13)
40
CALL QFLUX
9999
IPR=1
IPR= IPR+1
IPR≤8
35
COMPUTE AMPS(ISL,IPR) POWER
KP
TPOWER=0.0
ISTATE
ISTATE= ISTATE+1
ISTATE≤ 13
CALL TLIQD
1
COMPUTE TPOWER

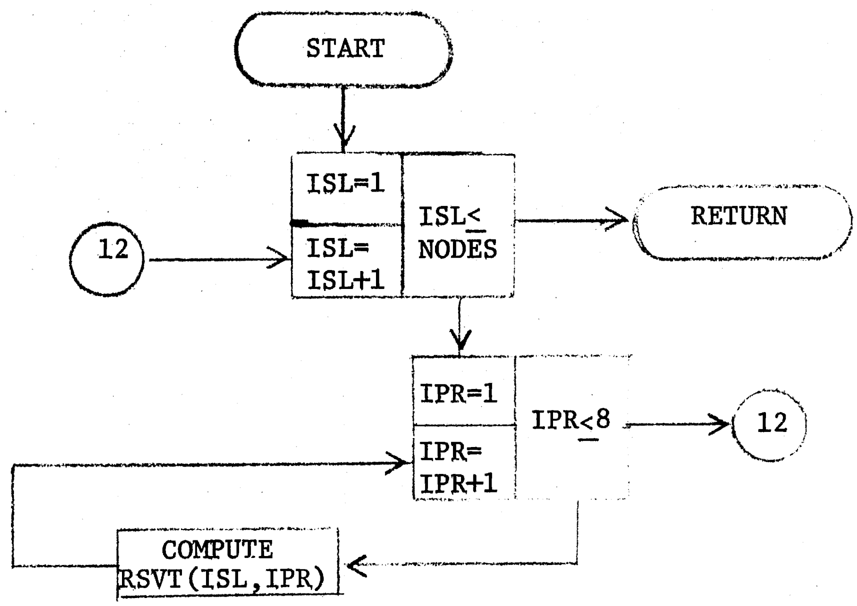
```
C      MAIN PROGRAM
       DIMENSION TORIG(2),TROOMC(2),TSAVE1(11,8),TSAVE2(11,8)
      1 , TCHCK1(8),TCHCK2(8)
       COMMON/READ1/TROOM,TCYAVG,PATM,VOLTS,TAMPS,MS,MW,NRUN
       COMMON/READ3/TOSURF(11,8),TISURF(11,8)
       COMMON/TCOND/CONDK(11,8)
       COMMON /TEMP1/ TWALL(11,8),AMPS(11,8),RESIS(11,8),POWERS(13)
      1,TPOWER
       COMMON /THERM1/TSAT(13),TSTART,TEND,QLOSS(2)
       COMMON/ERESIS/RSVTY(11,8)
       COMMON/MAIN1/IST,KOUNT
       COMMON /MAIN2/ AMP,OHMS,OHMS12,OHMS13
       COMMON /GEOM1/ XAREA(11,8),R(11   ),L(13,8),LTC(11),LEND(2),DELTAR(
      18),R456(11,8),LIN(2),LOUT(2),LTOTAL(2)
       COMMON/GEOM2/RBENDS,RBENDL,DOUT,DIN,PHI,DPHI,DELR,NODES,NSLICE
       REAL*4 L,LEND,LTC,MS,MW,LIN,LOUT,LTOTAL
     1 CONTINUE
       READ(5,1010) NRUN
       IF (NRUN.EQ.0)GO TO 3333
  1010 FORMAT(I3)
       CALL READS
       ITSECT=1
       IF(NRUN.GT.199)ITSECT=2
       CALL GEOM
       CALL CORECT
       NNODE=NODES-1
       TORIG(1)=211.5
       TORIG(2)=210.3
       TROOMC(1)=78.2
       TROOMC(2)=76.0
C  START SOLUTION WITH STATION 1
       DO 9999 IST=1,11
       DO 9 IPR=1,8
     9 TCHCK1(IPR) =0.0
C  SET ALL RADIAL TEMPERATURES EQUAL TO THE OUTSIDE SURFACE TEMPERATURES
       DO 10 ISL=1,NODES
       DO 10 IPR=1,8
    10 TWALL(ISL,IPR)=TOSURF(IST,IPR)
C  CALCULATE THERMAL CONDUCTIVITY FOR EACH NODE
       KOUNT=1
    19 CONTINUE
       CALL THCOND
C  CALCULATE ELECTRICAL CONDUCTIVITY FOR EACH NODE
       CALL ERSTVT
C  CALCULATE RESISTANCE & CURRENT FOR EACH SEGMENT
C  CALCULATE EQUIVALENT RESISTANCE FOR PARALLEL CIRCUITS
       IF(IST.LT.4.OR.IST.GT.6) CALL GEOMST
       IF(IST.EQ.4.OR.IST.EQ.5.OR.IST.EQ.6) CALL GEOMBT
       RINV = 0.0
       DO 11 ISL=1,NODES
       DO 11 IPR=1,8
       RESIS(ISL,IPR) = RSVTY(ISL,IPR)*L(IST,IPR)/XAREA(ISL,IPR)
       RINV = RINV +1.0/RESIS(ISL,IPR)
    11 CONTINUE
       OHMS = 1.0/RINV
C  CALCULATE CURRENT FOR EACH SEGMENT
       AMP=0.0
       DO 12 ISL=1,NODES
```

```
FORTRAN IV G LEVEL  21                MAIN                DATE = 75141          13/33/32

0048              DO 12 IPR=1,8
0049              AMPS(ISL,IPR) = TAMPS*OHMS/RESIS(ISL,IPR)
0050              AMP=AMP+AMPS(ISL,IPR)
0051           12 CONTINUE
            C  CALCULATE TEMPERATURES AT NODE 2
0052              ISL=1
0053              DO 13 IPR=1,8
0054              IMINS=IPR-1
0055              IPLUS=IPR+1
0056              NMINS = ISL - 1
0057              NPLUS = ISL + 1
0058              IF(IMINS.EQ.0) IMINS=8
0059              IF(IPLUS.EQ.9) IPLUS=1
0060              A= AMPS(ISL,IPR)*AMPS(ISL,IPR)*3.41213/XAREA(ISL,IPR)
0061              B= QLOSS(ITSECT)/(LEND(ITSECT)* (TORIG(ITSECT)-TROOMC(ITSECT))*8.)
0062              IF(IST.EQ.4.OR.IST.EQ.5.OR.IST.EQ.6) GO TO 101
0063              C=DELR/(2.0*24.0*DPHI)*(CONDK(ISL,IPR)+CONDK(ISL,IPLUS))/R(ISL)
0064              D= DELR/(2.0*24.0*DPHI)* (CONDK(ISL,IPR)+CONDK(ISL,IMINS))/R(ISL)
0065              X= DPHI/(24.0*DELR)*(CONDK(ISL,IPR)+CONDK(NPLUS,IPR))*(R( NPLUS)+
                 1DELR/2.0)
0066              GO TO 102
0067          101 CONTINUE
0068              C = DELTAR(IPR)/(2.0*24.0*DPHI)*(CONDK( ISL,IPR)+CONDK(ISL,IPLUS))/
                 1R456(ISL,IPR)
0069              D = DELTAR(IPR)/(2.0*24.0*DPHI)*(CONDK(ISL,IPR)+CONDK(ISL,IMINS))/
                 1R456(ISL,IPR)
0070              X =DPHI/(24.0*DELTAR(IPR))*(CONDK(ISL,IPR)+CONDK(NPLUS,IPR))*(R456
                 1(NPLUS,IPR)+DELTAR(IPR)/2.0)
0071          102 CONTINUE
0072              TWALL(NPLUS,IPR) = TWALL(ISL,IPR)- (A*RSVTY(ISL,IPR)+B*(TROOM-TWAL
                 1L(ISL,IPR))+ C*(TWALL(ISL,IPLUS)-TWALL(ISL,IPR))+D*(TWALL(ISL,IMIN
                 2S)-TWALL(ISL,IPR)))/X
0073           13 CONTINUE
            C  CALCULATE REMAINING NODAL TEMPERATURES
0074              DO 14 ISL=2,NNODE
0075              DO 14 IPR=1,8
0076              IMINS=IPR-1
0077              IPLUS=IPR+1
0078              NMINS=ISL-1
0079              NPLUS=ISL+1
0080              IF(IMINS.EQ.0) IMINS= 8
0081              IF(IPLUS.EQ.9) IPLUS = 1
0082              A = 3.41213*AMPS (ISL,IPR)*AMPS(ISL,IPR)/ XAREA(ISL,IPR)
0083              IF (IST.EQ.4.OR.IST.EQ.5.OR.IST.EQ.6) GO TO 103
0084              B = DPHI/(24.0*DELR)*(CONDK(ISL,IPR)+CONDK(NMINS,IPR))*(R(NMINS)
                 1-DELR/2.0)
0085              C = DELR/(24.0*DPHI)*(CONDK(ISL,IPR)+CONDK(ISL,IPLUS))/ R(ISL)
0086              D = DELR/(24.0*DPHI)*(CONDK(ISL,IPR)+CONDK(ISL,IMINS))/ R(ISL)
0087              X = DPHI/(24.0*DELR)*(CONDK(ISL,IPR)+CONDK(NPLUS,IPR))*(R(NPLUS)
                 1+DELR/2.0)
0088              GO TO 104
0089          103 CONTINUE
0090              B =DPHI/(24.0*DELTAR(IPR))*(CONDK(ISL,IPR)+CONDK(NMINS,IPR))*(R456
                 1(NMINS,IPR)-DELTAR(IPR)/2.0)
0091              C =DELTAR(IPR)/(24.0*DPHI)*(CONDK(ISL,IPR)+CONDK(ISL,IPLUS))/R456(
                 1ISL,IPR)
0092              D =DELTAR(IPR)/(24.0*DPHI)*(CONDK(ISL,IPR)+CONDK(ISL,IMINS))/R456(
                 1ISL,IPR)
```

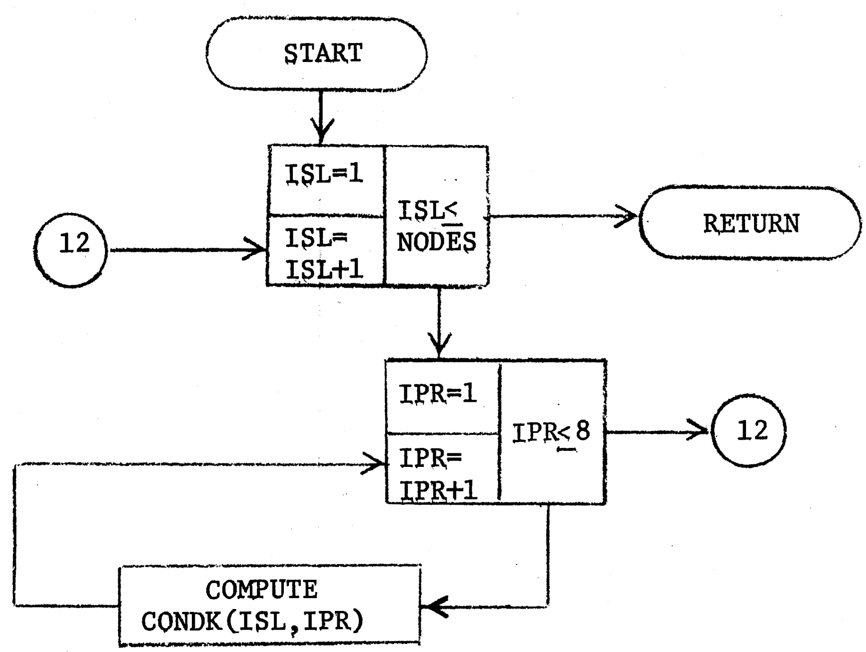
```
            X =DPHI/(24.0*DELTAR(IPR))*(CONDK(ISL,IPR)+CONDK(NPLUS,IPR))*(R456
           1(NPLUS,IPR)+DELTAR(IPR)/2.0)
      104 CONTINUE
            TWALL(NPLUS,IPR)= TWALL(ISL,IPR) - (A*RSVTY(ISL,IPR)+B*(TWALL(NMIN
           1S,IPR)-TWALL(ISL,IPR)) +C* (TWALL(ISL,IPLUS)-TWALL(ISL,IPR))+ D*(T
           2WALL(ISL,IMINS)-TWALL(ISL,IPR)))/X
       14 CONTINUE
          DO 15 IPR=1,8
          TCHCK2(IPR) = TWALL(NODES,IPR)
       15 CONTINUE
          DO 16 IPR =1,8
          IF (ABS(TCHCK2(IPR)-TCHCK1(IPR)).GT.0.005) GO TO 17
       16 CONTINUE
          GO TO 20
       17 DO 18 IPR=1,8
       18 TCHCK1(IPR) = TCHCK2(IPR)
          KOUNT = KOUNT+1
          GO TO 19
       20 DO 21 IPR=1,8
          TISURF( IST ,IPR)=TWALL(NODES,IPR)
       21 CONTINUE
    C  CALCULATE POWER GENERATED IN EACH SEGMENT IN BTU/HOUR
          POWER =0.0
          DO 22 ISL=1,NODES
          DO 22 IPR=1,8
          POWER=POWER+AMPS(ISL,IPR)*AMPS(ISL,IPR)*RESIS(ISL,IPR)
       22 CONTINUE
          POWERS(IST)=POWER*3.41213
    C  CALCULATE POWER GENERATED IN SEGMENTS 12 & 13 BY SAVING VARIABLES FOR
    C  STATIONS 1,2,10,AND 11
          IF(IST.GT.2) GO TO 23
          IF(IST.EQ.2) GO TO 24
          DO 25 ISL=1,NODES
          DO 25 IPR=1,8
       25 TSAVE1(ISL,IPR)=TWALL(ISL,IPR)
          GO TO 23
       24 DO 26 ISL=1,NODES
          DO 26 IPR=1,8
       26 TSAVE2(ISL,IPR)=TWALL(ISL,IPR)
          DO 27 ISL=1,NODES
          DO 27 IPR=1,8
       27 TWALL(ISL,IPR)=(TSAVE1(ISL,IPR)+TSAVE2(ISL,IPR))/2.0
          CALL ERSTVT
          RINV=0.0
          DO 28 ISL=1,NODES
          DO 28 IPR=1,8
          RESIS(ISL,IPR) = RSVTY(ISL,IPR)*L( 12,IPR)/XAREA(ISL,IPR)
          RINV=RINV+1.0/RESIS(ISL,IPR)
    C  REDEFINE WALL TEMPERATURES AT STATION 2
          TWALL(ISL,IPR) = TSAVE2(ISL,IPR)
       28 CONTINUE
          OHMS12=1.0/RINV
          POWER=0.0
          DO 29 ISL=1,NODES
          DO 29 IPR=1,8
          AMPS(ISL,IPR)= TAMPS*OHMS12/RESIS(ISL,IPR)
       29 POWER=POWER +AMPS(ISL,IPR)*AMPS(ISL,IPR)*RESIS(ISL,IPR)
          POWERS(12)=POWER*3.41213
```

```
   23 IF(IST.LT.10) GO TO 40
      IF(IST.EQ.11) GO TO 30
      DO 31 ISL=1,NODES
      DO 31 IPR=1,8
   31 TSAVE1(ISL,IPR)=TWALL(ISL,IPR)
      GO TO 40
   30 DO 32 ISL=1,NODES
      DO 32 IPR=1,8
   32 TSAVE2(ISL,IPR)=TWALL(ISL,IPR)
      DO 33 ISL=1,NODES
      DO 33 IPR=1,8
   33 TWALL(ISL,IPR)=(TSAVE1(ISL,IPR)+TSAVE2(ISL,IPR))/2.0
      CALL ERSTVT
      RINV=0.0
      DO 34 ISL=1,NODES
      DO 34 IPR=1,8
      RESIS(ISL,IPR) = RSVTY(ISL,IPR)*L( 13,IPR)/XAREA(ISL,IPR)
      RINV=RINV+1.0/RESIS(ISL,IPR)
C  REDEFINE WALL TEMPERATURES AT STATION 10
      TWALL(ISL,IPR) = TSAVE2(ISL,IPR)
   34 CONTINUE
      OHMS13=1.0/RINV
      POWER=0.0
      DO 35 ISL=1,NODES
      DO 35 IPR=1,8
      AMPS(ISL,IPR)= TAMPS*OHMS13/RESIS(ISL,IPR)
   35 POWER=POWER + AMPS(ISL,IPR)*AMPS(ISL,IPR)*RESIS(ISL,IPR)
      POWERS(13)=POWER*3.41213
   40 CONTINUE
      CALL QFLUX
 9999 CONTINUE
C  CALCULATE TOTAL POWER GENERATED IN BTU/HOUR
      TPOWER=0.0
      DO 61 ISTAT=1,13
   61 TPOWER=TPOWER+POWERS(ISTAT)
      CALL TLIQD
      GO TO 1
 3333 STOP
      END
```

SUBROUTINE ERSTVT



SUBROUTINE THCOND



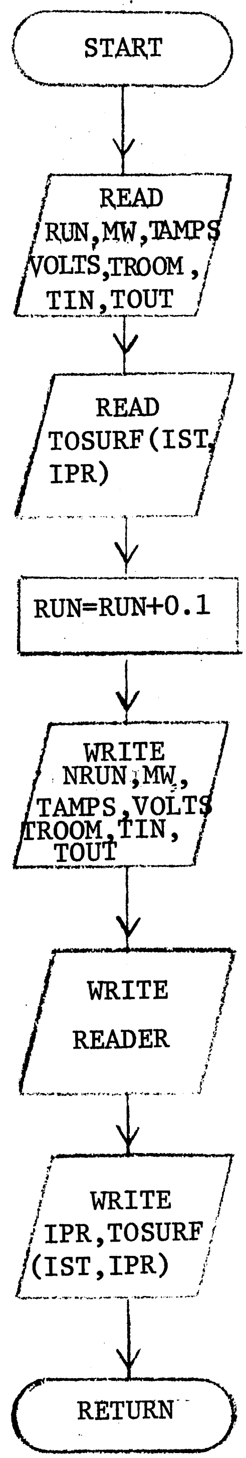
FORTRAN IV G LEVEL 21 ERSTVT DATE = 75141 13/33/32

```
      SUBROUTINE ERSTVT
      COMMON/ERESIS/RSVTY(11,8)
      COMMON /TEMP1/ TWALL(11,8),AMPS(11,8),RESIS(11,8),POWERS(13)
     1,TPOWER
      COMMON/GEOM2/RBENDS,RBENDL,DOUT,DIN,PHI,DPHI,DELR,NODES,NSLICE
C ELECTRICAL RESISTIVITY OF INCONEL 600 IN OHMS-SQIN/IN
      DO 12 ISL=1,NODES
      DO 12 IPR=1,8
      RSVTY(ISL,IPR)=(40.40292+2.515385E-3*TWALL(ISL,IPR))*1.0E-6
   12 CONTINUE
      RETURN
      END
```

FORTRAN IV G LEVEL 21 THCOND DATE = 75141 13/33/32

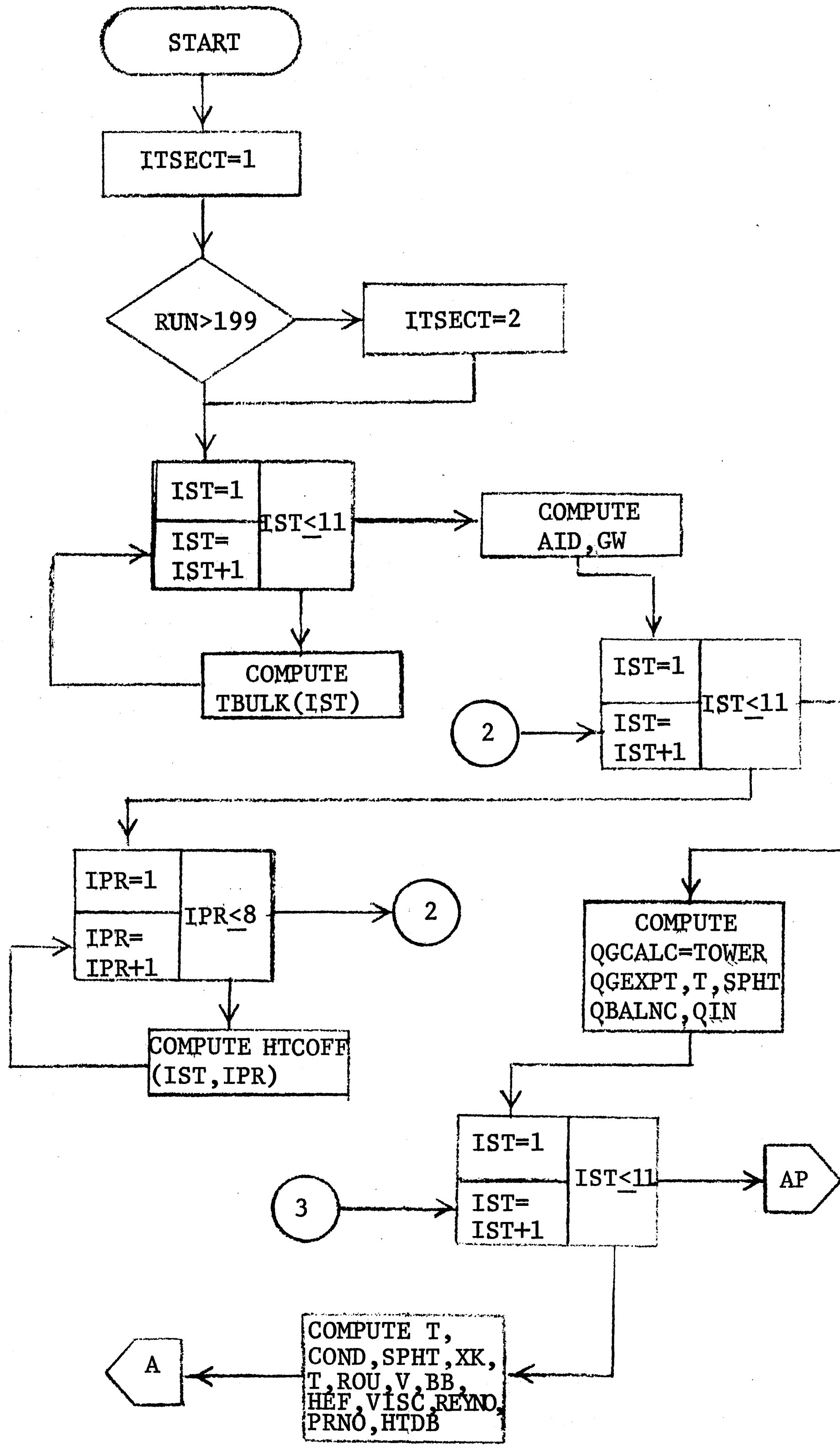
```
      SUBROUTINE THCOND
      COMMON/TCOND/CONDK(11,8)
      COMMON /TEMP1/ TWALL(11,8),AMPS(11,8),RESIS(11,8),POWERS(13)
     1,TPOWER
      COMMON/GEOM2/RBENDS,RBENDL,DOUT,DIN,PHI,DPHI,DELR,NODES,NSLICE
C THERMAL CONDUCTIVITY OF INCONEL 600 IN BTU/(HR-FT-DEGF)
      DO 12 ISL=1,NODES
      DO 12 IPR=1,8
      CONDK(ISL,IPR) =8.313769+3.846154E-3*TWALL(ISL,IPR)
   12 CONTINUE
      RETURN
      END
```

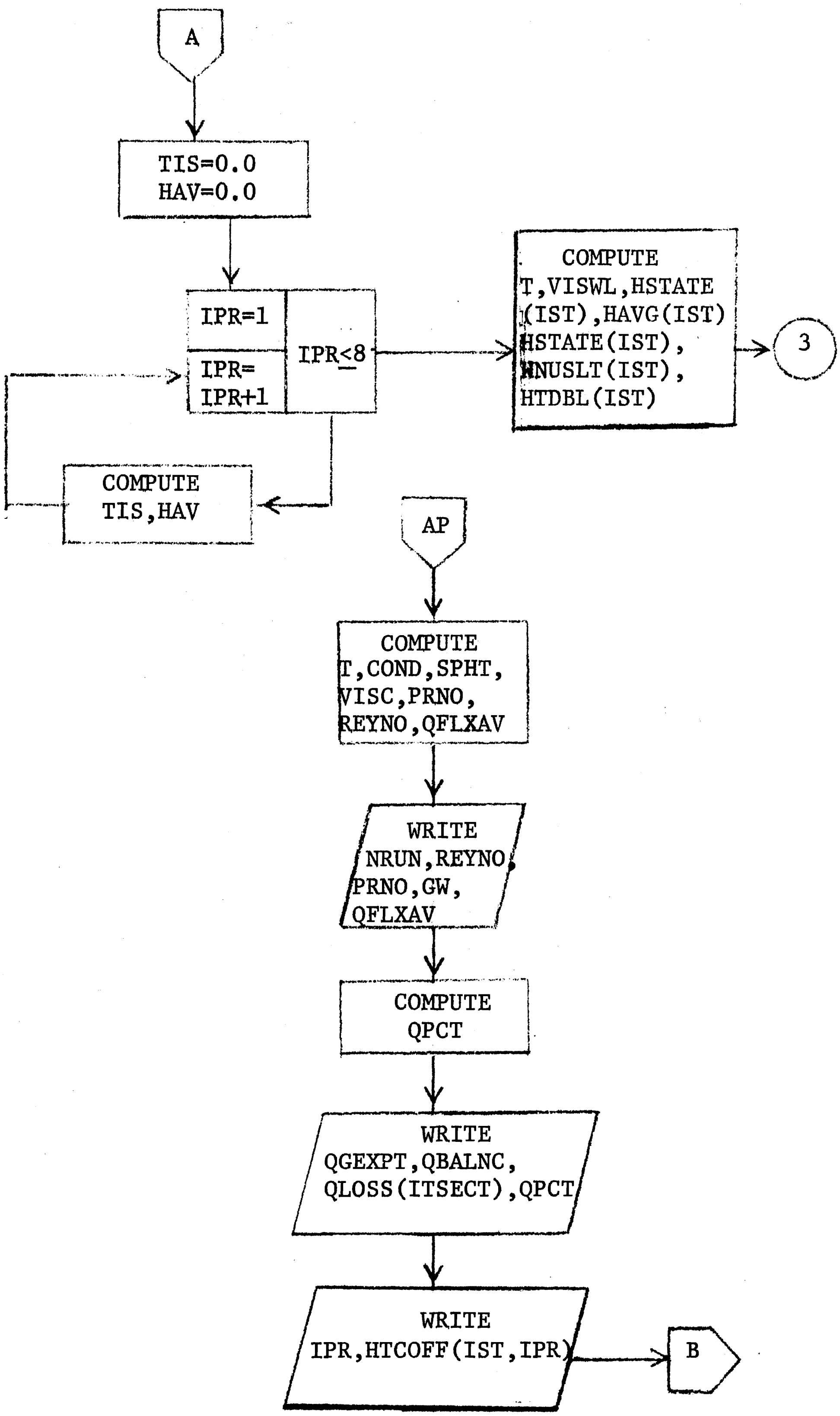
SUBROUTINE READ

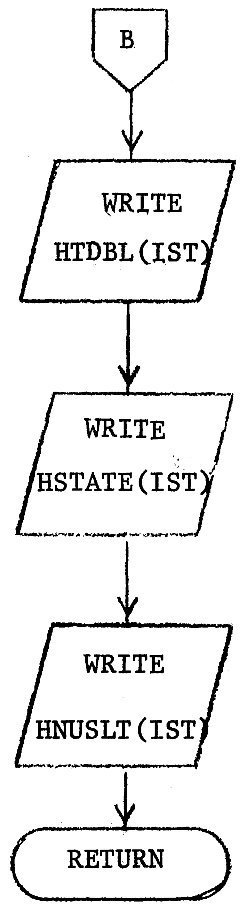


```
FORTRAN IV G LEVEL  21                 READS                DATE = 75141          13/33/32

 0001              SUBROUTINE READS
 0002              COMMON/READ1/TROOM,TCYAVG,PATM,VOLTS,TAMPS,MS,MW,NRJN
 0003              COMMON/READ2/TIN,TOUT,PIN,POUT,P1,P2,P3,P4
 0004              COMMON/READ3/TOSURF(11,8),TISURF(11,8)
 0005              REAL*4 MW
 0006              READ (5,1)            RUN,MW,TAMPS,VOLTS,TROOM,TIN,TOUT
 0007            1 FORMAT(7F10.3)
 0008              READ (5,2)((TOSURF(IST,IPR),IPR=1,8),IST=1,11)
 0009            2  FORMAT(8F10.1)
 0010              NRUN = RUN +0.1
 0011              WRITE(6,101) NRUN,MW,TAMPS,VOLTS,TROOM,TIN,TOUT
 0012          101 FORMAT(33X,15('-')/33X,'RUN NUMBER   ',I3/33X,15('-')//
                 1       15X,' WATER FLOW RATE                   =',F8.2,2X,'LBM/HOUR'/
                 2       15X,'CURRENT TO TUBE                    =',F8.2,2X,' AMPS'/
                 3       15X,'VOLTAGE DROP IN TUBE               =',F8.2,2X,'VOLTS'/
                 4       15X,'ROOM TEMPERATURE                   =',F8.2,2X,'DEGREES F'/
                 5       15X,'UNCORRECTED INLET TEMPERATURE      =',F8.2,2X,'DEGREES F'/
                 6       15X,'UNCORRECTED OUTLET TEMPERATURE     =',F8.2,2X,'DEGREES F'/
                 7)
 0013              WRITE(6,102)
 0014          102  FORMAT(///20X,'   OUTSIDE SURFACE TEMPERATURES - DEGREES F'///
                 1 9X,'1',6X,'2',6X,'3',6X,'4',6X,'5',6X,'6',6X,'7',6X,'8',
                 26X,'9',6X,'10',5X,'11'//)
 0015              WRITE(6,103)(IPR,(TOSURF(IST,IPR),IST=1,11),IPR=1,8)
 0016          103 FORMAT(3X,I1,F8.1,10F7.1 )
 0017              RETURN
 0018              END
```

SUBROUTINE TLIQD




A
TIS=0.0
HAV=0.0
IPR=1
IPR=
IPR+1
IPR≤8
COMPUTE
T,VISWL,HSTATE
(IST),HAVG(IST)
HSTATE(IST),
NNUSLT(IST),
HTDBL(IST)
3
COMPUTE
TIS,HAV
AP
COMPUTE
T,COND,SPHT,
VISC,PRNO,
REYNO,QFLXAV
WRITE
NRUN,REYNO,
PRNO,GW,
QFLXAV
COMPUTE
QPCT
WRITE
QGEXPT,QBALNC,
QLOSS(ITSECT),QPCT
WRITE
IPR,HTCOFF(IST,IPR)
B

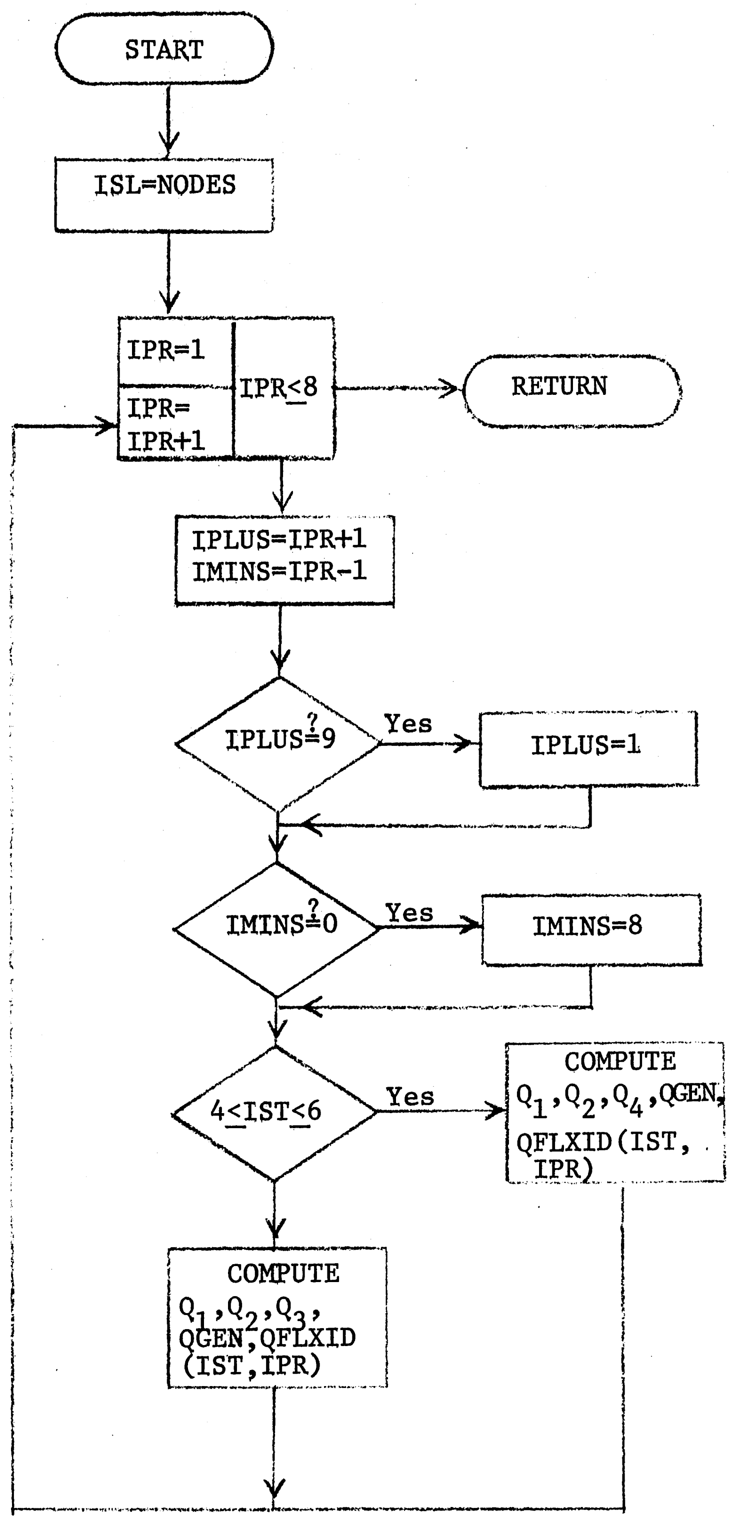

B
WRITE
HTDBL(IST)
WRITE
HSTATE(IST)
WRITE
HNUSLT(IST)
RETURN

```
      SUBROUTINE  TLIQD
      COMMON/READ1/TROOM,TCYAVG,PATM,VOLTS,TAMPS,MS,MW,NRUN
      COMMON/READ2/TIN,TOUT,PIN,POUT,P1,P2,P3,P4
      COMMON/READ3/TOSURF(11,8),TISURF(11,8)
      COMMON /TEMP1/ TWALL(11,8),AMPS(11,8),RESIS(11,8),POWERS(13)
     1,TPOWER
      COMMON /GEOM1/ XAREA(11,8),R(11   ),L(13,8),LTC(11),LEND(2),DELTAR(
     18),R456(11,8),LIN(2),LOUT(2),LTOTAL(2)
      COMMON/GEOM2/RBENDS,RBENDL,DOUT,DIN,PHI,DPHI,DELR,NODES,NSLICE
      COMMON /THERM1/TSAT(13),TSTART,TEND,QLOSS(2)
      COMMON /THERM2/HTCOFF(11,8),QUALTY(13),HSTART,HEND,XSTART,XEND,ENT
     1H(13)
      COMMON /QFLUX1/ QFLXID(11,8)
      COMMON /TLIQ1/ TBULK(13),HLIQ(13)
      COMMON /TLIQ2/ HTDBL(11),HNUSLT(11),HSTATE(11) ,HAVG(11)
      DOUBLE PRECISION T,P
      REAL*4 LIN,LOUT,LTOTAL,L,LTC,LEND,MW,MS,MTOT
      ITSECT=1
      IF(NRUN.GT.199) ITSECT=2
C  CALCULATION OF FLUID BULK TEMPERATURE AT EACH STATION,DEG.F
      DO 1 IST=1,11
      TBULK(IST) = TIN+(TOUT-TIN)*LTC( IST   )/LEND(ITSECT)
    1 CONTINUE
      AID=3.1416*DIN*DIN/4.0/144.0
      GW=MW/AID
C  CALCULATION OF PERIPHERAL HEAT TRANSFER COEFFICIENT FROM EXPERIMENTAL
C  DATA,BTU/(HR-SQ.FT-DEG.F)
      DO 2 IST=1,11
      DO 2 IPR=1,8
      HTCOFF(IST,IPR) =QFLXID(IST,IPR)/(TISURF(IST,IPR)-TBULK(IST))
    2 CONTINUE
C  CALCULATION OF INPUT AND OUTPUT HEAT TRANSFER RATE,BTU/HR.
      QGCALC=TPOWER
      QGEXPT =TAMPS*VOLTS*3.41213
      T=(TOUT+TIN)/2.0
      SPHT=1.01881-0.4802E-3*T+0.3274E-5*T**2-0.604E-8*T**3
      QBALNC=MW*SPHT*(TOUT-TIN)
      QIN=QGCALC-QLOSS(ITSECT)
      DO 3 IST=1,11
      T=TBULK(IST)
      COND=0.30289+0.7029E-3*T-0.1178E-5*T**2-0.55E-9*T**3
      SPHT=1.01881-0.4802E-3*T+0.3274E-5*T**2-0.604E-8*T**3
C  EAGLE FERGUSON CHART CHART HEAT TRANSFER COEFFICIENT,
C  BTU/HR-SQ.FT-DEG.F
      XK=1.01E-3*T+0.919
      T=10.0*(T-32.0)/18.0
      ROU=62.37*(0.999986+0.189E-4*T+0.5886E-5*T**2+0.1548E-7*T**3)
      V=GW/ROU/3600.0
      BB=5.653*XK
      HEF=EXP(BB+0.805*ALOG(V))
      VISC=2.420*1.002*10.0**((1.3272*(20.0-T)-0.001053*(T-20.0)**2)/(T+
     1105.0))
      REYNO=GW*DIN/12.0/VISC
      PRNO=VISC*SPHT/COND
C  DITTUS-BOETLER HEAT TRANSFER COEFFICIENT,BTU/(HR-SQ.FT-DEG.F)
      HTDB=0.023*REYNO**0.8*PRNO**0.4*COND/DIN*12.0
      TIS=0.0
C  AVERAGE HEAT TRANSFER COEFFICIENT AT EACH STATION FROM EXPERIMENTAL
```

```
              C  DATA,BTU/HR-SQ.FT-DEG.F
 0047               HAV=0.0
 0048               DO 4 IPR=1,8
 0049               TIS=TIS+TISURF(IST,IPR)
 0050             4 HAV       =HAV+HTCOFF(IST,IPR)
              C  SIEDER-TATE HEAT TRANSFER COEFFICIENT AT EACH STATION,
              C  BTU/HR-SQ.FT-DEG.F
 C051               T=TIS/8.0
 0052               T=10.0*(T-32.0)/18.0
 0053               VISWL=2.42*1.002*10.0**((1.3272*(20.0-T)-0.001053*(T-20.0)**2)/(T+
                   1105.0))
 0054               HSTATE(IST)=0.027*REYNO**0.8*PRNO**0.333*(VISC/VISWL)**0.14*COND/
                   1DIN*12.0
 0055               HAVG(IST)=HAV/8.0
              C  RATIO OF HEAT TRANSFER COEFFICIENTS:THIS WORK TO SIEDER-TATE,
              C  EAGLE-FERGUSON CHART,DITTUS-BOETLER.
 0056               HSTATE(IST)=HAVG(IST)/HSTATE(IST)
 0057               HNUSLT(IST)=HAVG(IST)/HEF
 0058               HTDBL(IST)=HAVG(IST)/HTDB
 0059             3 CONTINUE
 0060               WRITE (6,100)
 0061           100 FORMAT(1H1)
 0062               T=(TIN+TOUT)/2.0
 0063               COND=0.30289+0.7029E-3*T-0.1178E-5*T**2-0.55E-9*T**3
 0064               SPHT=1.01881-0.4802E-3*T+0.3274E-5*T**2-0.604E-8*T**3
 0065               T=10.0*(T-32.0)/18.0
 0066               VISC=2.420*1.002*10.0**((1.3272*(20.0-T)-0.001053*(T-20.0)**2)/(T+
                   1105.0))
 0067               PRNO=VISC*SPHT/COND
 0068               REYNO=GW*DIN/12.0/VISC
 0069               QFLXAV=QIN/(3.1416*DIN/12.0*LEND(ITSECT)/12.0)
 0070               WRITE(6,112)NRUN,REYNO,PRNO,GW,QFLXAV
 0071           112 FORMAT(33X,15('-')/33X,'RUN NUMBER  ',I3/33X,15('-')//
                   6      10X,'AVERAGE REYNOLDS NUMBER          =',F9.2/
                   1      10X,'AVERAGE PRANDTL NUMBER           =',F9.2/
                   2      10X,'MASS FLUX                        =',F9.2,2X,'LBM/(SQ.FT-
                   3HR)'/
                   4      10X,'AVERAGE HEAT FLUX                =',F9.2,2X,'BTU/(SQ.FT-
                   5HR)')
 0072               QPCT=(QGEXPT-QLOSS(ITSECT)-QBALNC)/(.5*(QGEXPT-QLOSS(ITSECT)+QBALN
                   9C))
 C073               QPCT=100.0*QPCT
 0074               WRITE(6,789)QGEXPT,QBALNC,QLOSS(ITSECT),QPCT
 0075           789 FORMAT(10X,'Q=AMP*VOLT                         =',F9.2,2X,'BTU/HR'/
                   110X,'Q=M*C*(T2-T1)                  =',F9.2,2X,'BTU/HR'/
                   310X,'HEAT LOST                      =',F9.2,2X,'BTU/HR'/
                   510X,'HEAT BALANCE ERROR PERCENT     =',F9.2)
 0076               WRITE(6,333)
 0077           333 FORMAT(/,15X,'PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-D
                   $EG.F)')
 0078               WRITE(6,8888)
 0079          8888 FORMAT(/,13X,'1',6X,'2',6X,'3',6X,'4',6X,'5',6X,'6',6X,'7',6X,'8',
                   16X,'9',6X,'10',5X,'11')
 0080               WRITE(6,12)(IPR,(HTCOFF(IST,IPR),IST=1,11),IPR=1,8)
 0081            12 FORMAT(3X,I2,4X,11F7.1)
 0082               WRITE(6,101)
 0083           101 FORMAT(/,15X,'AVERAGE HEAT TRANSFER COEFFICIENT-BTU/(SQ.FT.HR-DEG.
                   2F)')
```

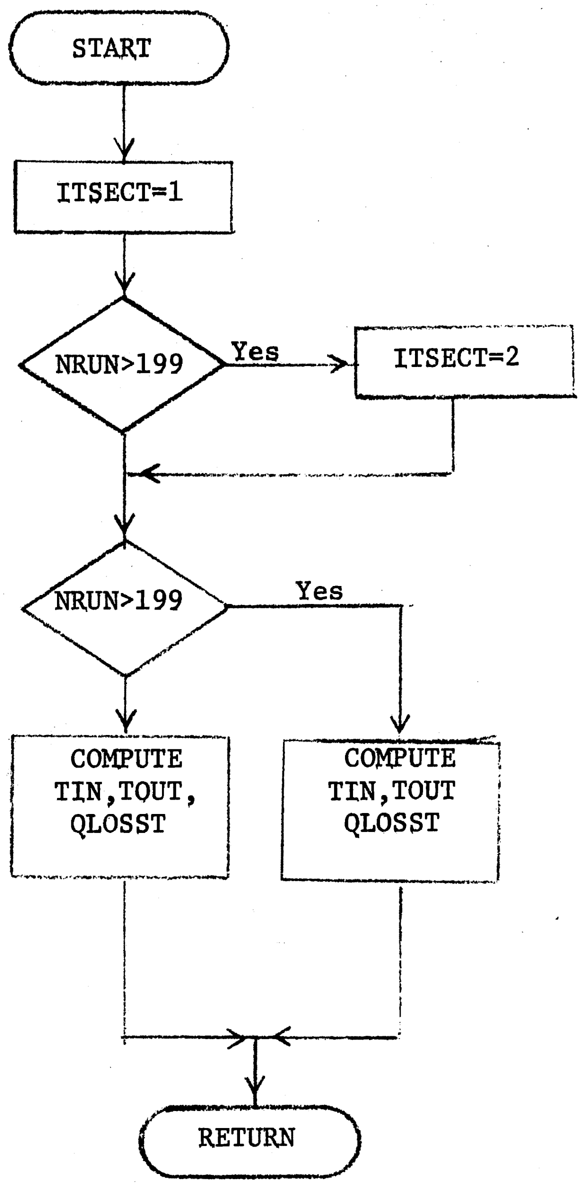
```
              WRITE(6,8888)
              WRITE(6,13)(HAVG(IST),IST=1,11)
           13 FORMAT(5X,'HAVG',11F7.1)
              WRITE(6,102)
          102 FORMAT(/6X,' RATIO OF CALCULATED HEAT TRANSFER COEFFICIENT TO THOS
             *E PREDICTED BY LITERATURE')
              WRITE(6,8888)
              WRITE(6,14)(HTDBL(IST),IST=1,11)
           14 FORMAT(4X,'DITBL',11F7.2)
              WRITE(6,15)(HSTATE(IST),IST=1,11)
           15 FORMAT(3X,'SIDTAT',11F7.2)
              WRITE(6,16)(HNUSLT(IST),IST=1,11)
           16 FORMAT(3X,'HEFCHT',11F7.2)
              WRITE(6,100)
              RETURN
              END
```

SUBROUTINE QFLUX



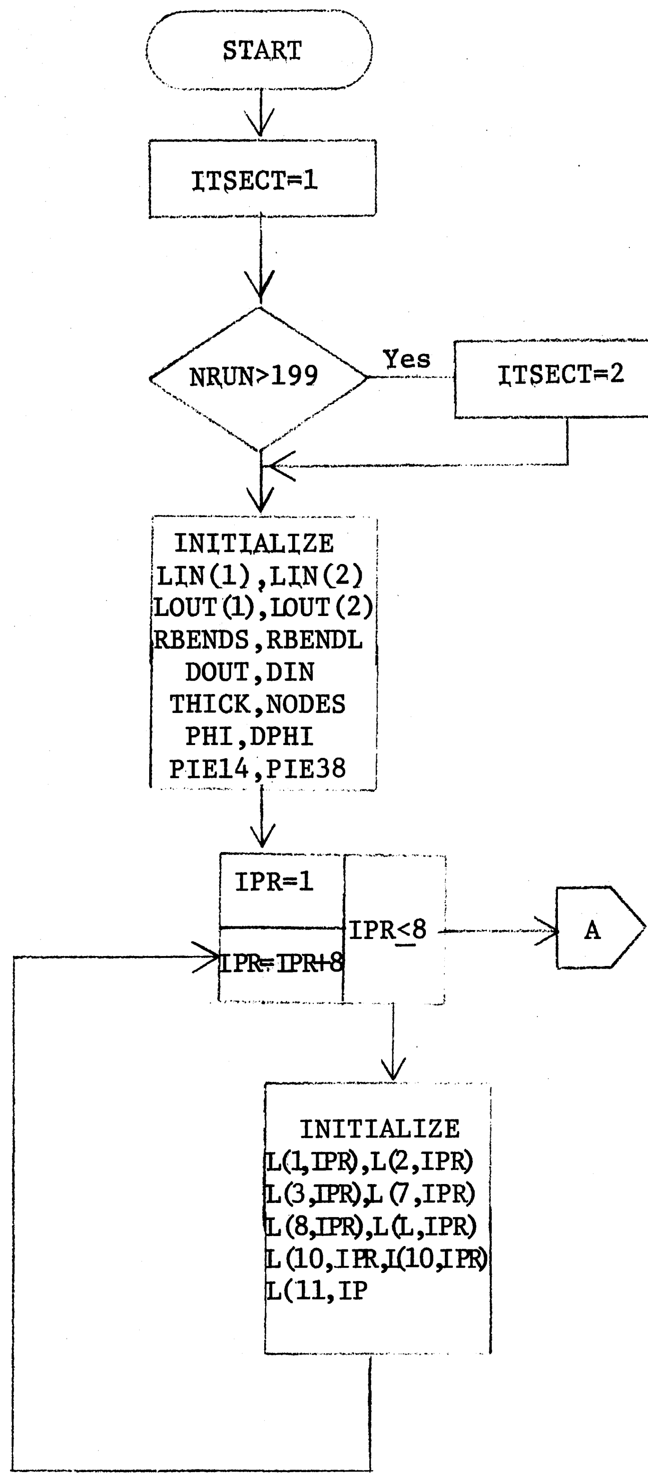
```
0001          SUBROUTINE QFLUX
0002          COMMON /MAIN1/ IST,KOUNT
0003          COMMON /GEOM1/ XAREA(11,8),R(11  ),L(13,8),LTC(11),LEND(2),DELTAR(
             18),R456(11,8),LIN(2),LOUT(2),LTOTAL(2)
0004          COMMON/GEOM2/RBENDS,RBENDL,DOUT,DIN,PHI,DPHI,DELR,NODES,NSLICE
0005          COMMON/TCOND/CONDK(11,8)
0006          COMMON /TEMP1/ TWALL(11,8),AMPS(11,8),RESIS(11,8),POWERS(13)
             1,TPOWER
0007          COMMON /QFLUX1/ QFLXID(11,8)
0008          COMMON/ERESIS/RSVTY(11,8)
0009          COMMON /QFLUX2/ Q1,Q2,Q4,QGEN
        C  CALCULATE HEAT FLUX AT INSIDE SURFACE BY MAKING A HEAT BALANCE
0010          ISL=NODES
0011          DO 10 IPR=1,8
0012          IPLUS=IPR+1
0013          IMINS=IPR-1
0014          IF (IPLUS.EQ.9) IPLUS=1
0015          IF (IMINS.EQ.0) IMINS=8
0016          IF (IST.EQ.4.OR.IST.EQ.5.OR.IST.EQ.6) GO TO 8
0017          Q1 = DPHI/(24.0*DELR)*(CONDK(ISL-1,IPR)+CONDK(ISL,IPR))*(R(ISL-1)
             1-DELR/2.0)*(TWALL(ISL-1,IPR)-TWALL(ISL,IPR))
0018          Q2 = DELR/(2.0*24.0*DPHI)/R(ISL)*(CONDK(ISL,IPLUS)+CONDK(ISL,IPR))
             1*(TWALL(ISL,IPLUS)-TWALL(ISL,IPR))
0019          Q4 = DELR/(2.0*24.0*DPHI)/R(ISL)*(CONDK(ISL,IMINS)+CONDK(ISL,IPR))
             1*(TWALL(ISL,IMINS)-TWALL(ISL,IPR))
0020          QGEN =3.41213*AMPS(ISL,IPR)*AMPS(ISL,IPR)*RSVTY(ISL,IPR)/XAREA(ISL
             1,IPR)
0021          QFLXID(IST,IPR) =(Q1+Q2+Q4+QGEN)/(R(NODES)*DPHI)*144.0
0022          GO TO 9
0023        8 CONTINUE
0024          Q1 = DPHI/(24.0*DELTAR(IPR))*(CONDK(ISL-1,IPR)+CONDK(ISL,IPR))*
             1(R456(ISL-1,IPR)-DELTAR(IPR)/2.0)* (TWALL(ISL-1,IPR)-TWALL(ISL,IPR
             2))
0025          Q2 = DELTAR(IPR)/(2.0*24.0*DPHI)/R456(ISL,IPR)*(CONDK(ISL,IPLUS)+
             1CONDK(ISL,IPR))*(TWALL(ISL,IPLUS)-TWALL(ISL,IPR))
0026          Q4 = DELTAR(IPR)/(2.0*24.0*DPHI)/R456(ISL,IPR)*(CONDK(ISL,IMINS)+
             1CONDK(ISL,IPR))*(TWALL(ISL,IMINS)-TWALL(ISL,IPR))
0027          QGEN =3.41213*AMPS(ISL,IPR)*AMPS(ISL,IPR)*RSVTY(ISL,IPR)/XAREA(ISL
             1,IPR)
0028          QFLXID(IST,IPR) =(Q1+Q2+Q4+QGEN)/(R456(NODES,IPR)*DPHI)*144.0
0029        9 CONTINUE
0030       10 CONTINUE
0031          RETURN
0032          END
```

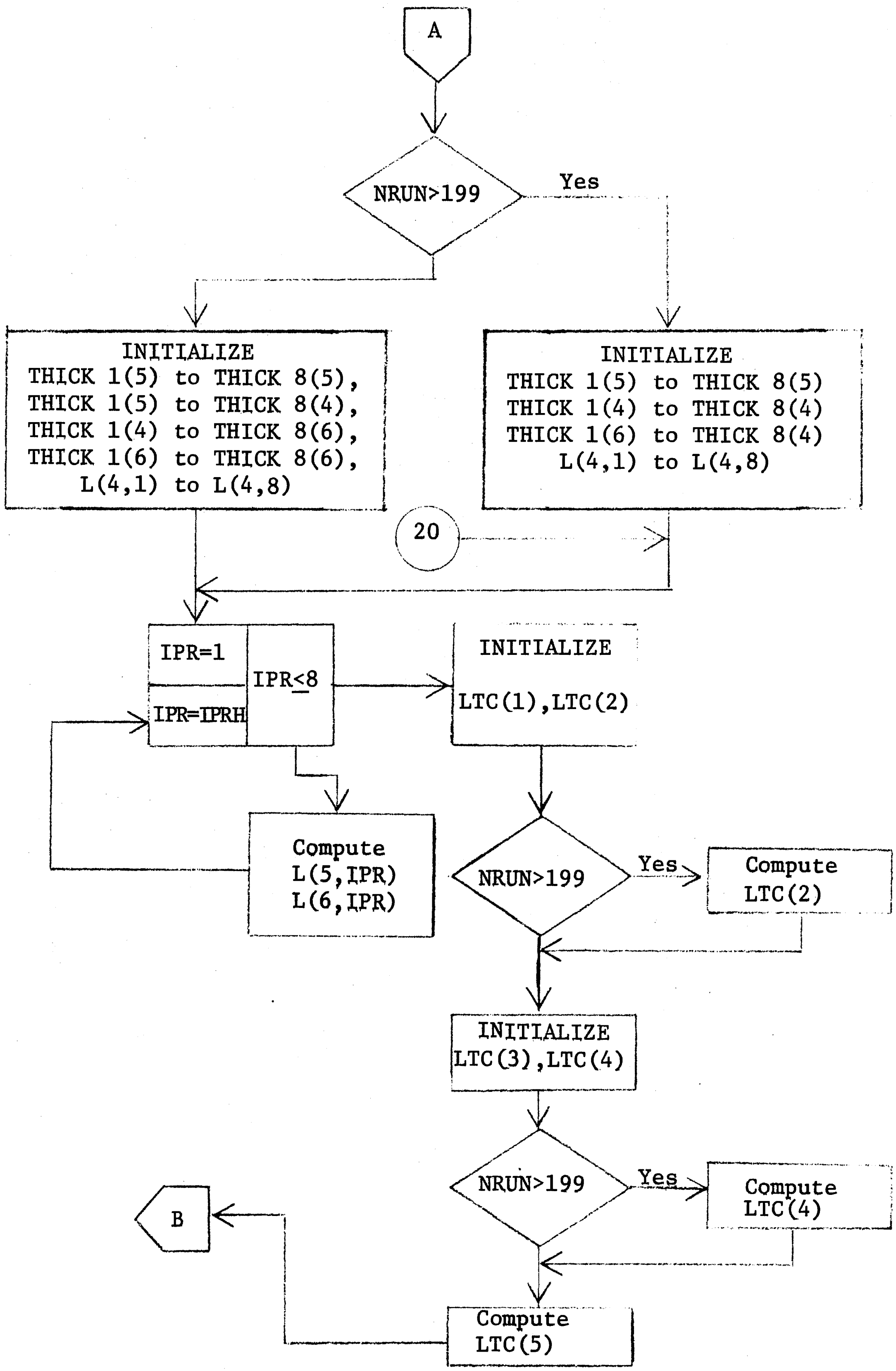
SUBROUTINE CORECT

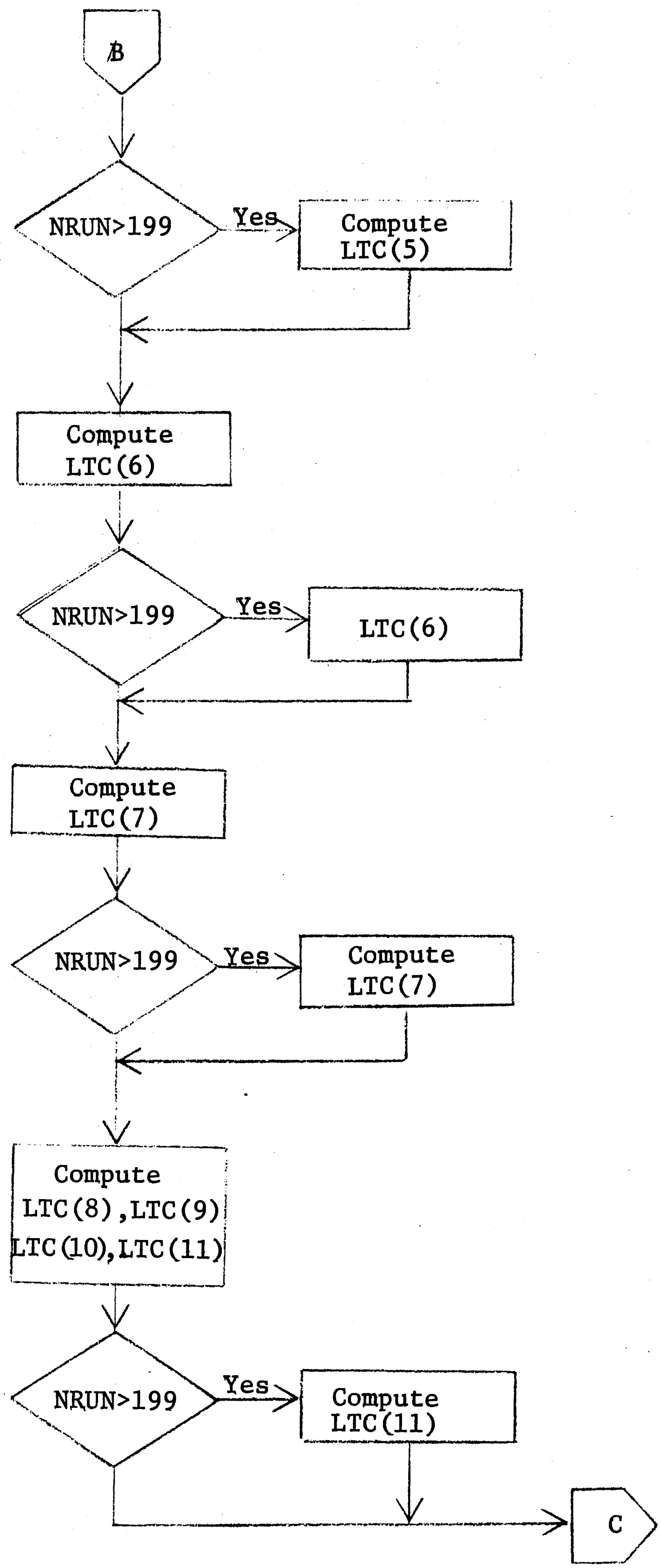


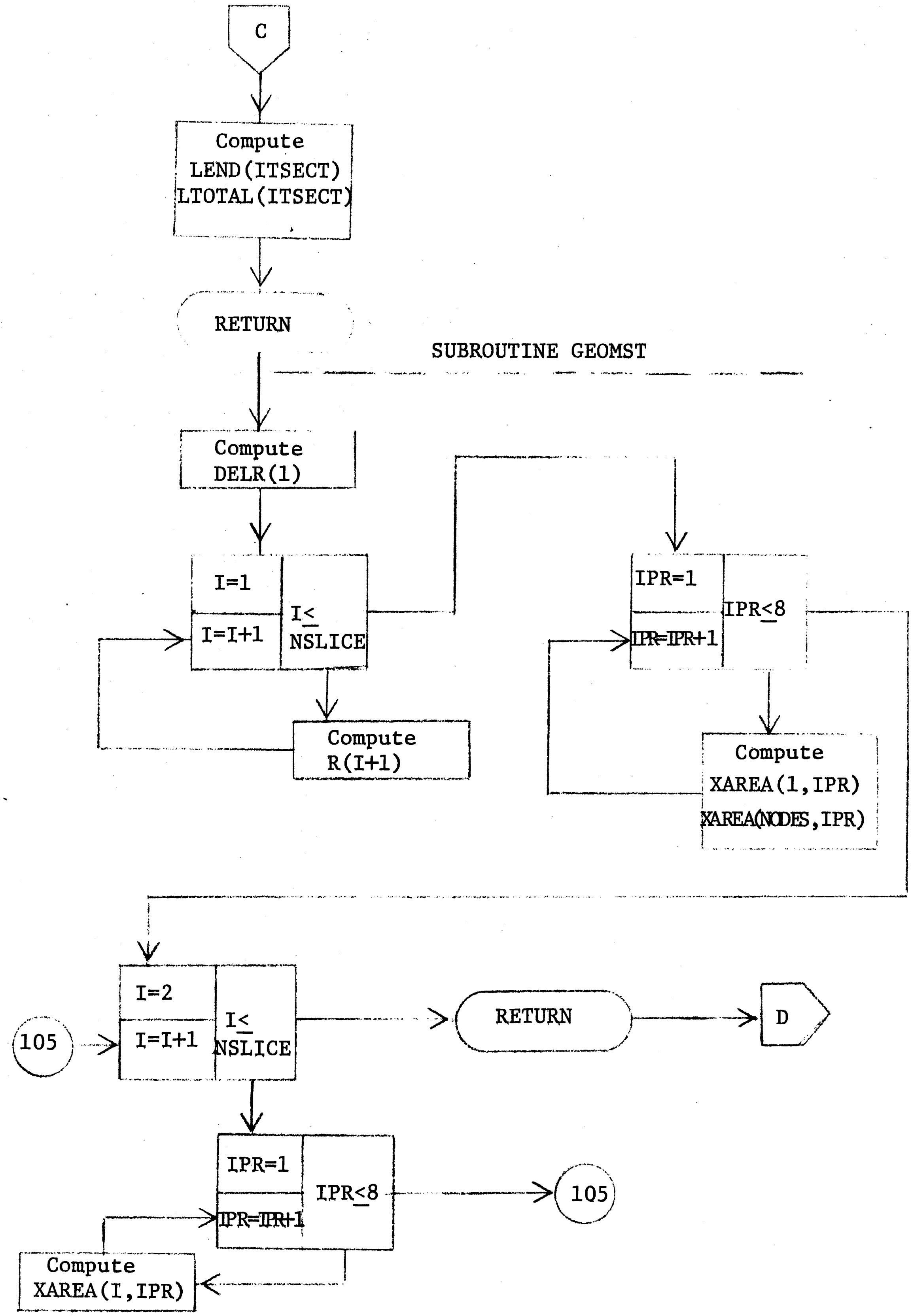
```
FORTRAN IV G LEVEL  21                  CORECT              DATE = 75141           13/33/32

                  SUBROUTINE CORECT
                  COMMON/READ1/TROOM,TCYAVG,PATM,VOLTS,TAMPS,MS,MW,NRUN
                  COMMON/READ2/TIN,TOUT,PIN,POUT,P1,P2,P3,P4
                  COMMON /GEOM1/ XAREA(11,8),R(11  ),L(13,8),LTC(11),LEND(2),DELTAR(
                 18),R456(11,8),LIN(2),LOUT(2),LTOTAL(2)
                  COMMON /THERM1/TSAT(13),TSTART,TEND,QLOSS(2)
                  REAL*4 LEND,LTOTAL
            C  CORRECT INLET AND OUTLET MIXTURE TEMPERATURES AND CALCULATE TEST
            C  SECTION HEAT LOSSES
                  ITSECT =1
                  IF (NRUN.GT.199) ITSECT=2
                  IF (NRUN.GT.199) GO TO 11
                  TIN = TIN -.7/(210.7-72.0)*(TIN-TROOM)
                  TOUT=TOUT-.9/(210.7-72.0)*(TOUT-TROOM)
                  QLOSST= 596.0/(211.5-78.2)*((TIN+TOUT)/2.0-TROOM)
                  GO TO 12
               11 CONTINUE
                  TIN = TIN -.7/(210.3-76.5)*(TIN-TROOM)
                  TOUT=TOUT -.9 /(211.1-76.5)*(TOUT-TROOM)
                  QLOSST= 485.0/(210.3-76.0)*((TIN+TOUT)/2.0-TROOM)
               12 CONTINUE
                  QLOSS(ITSECT)=QLOSST*LEND(ITSECT)/LTOTAL(ITSECT)
                  RETURN
                  END
```

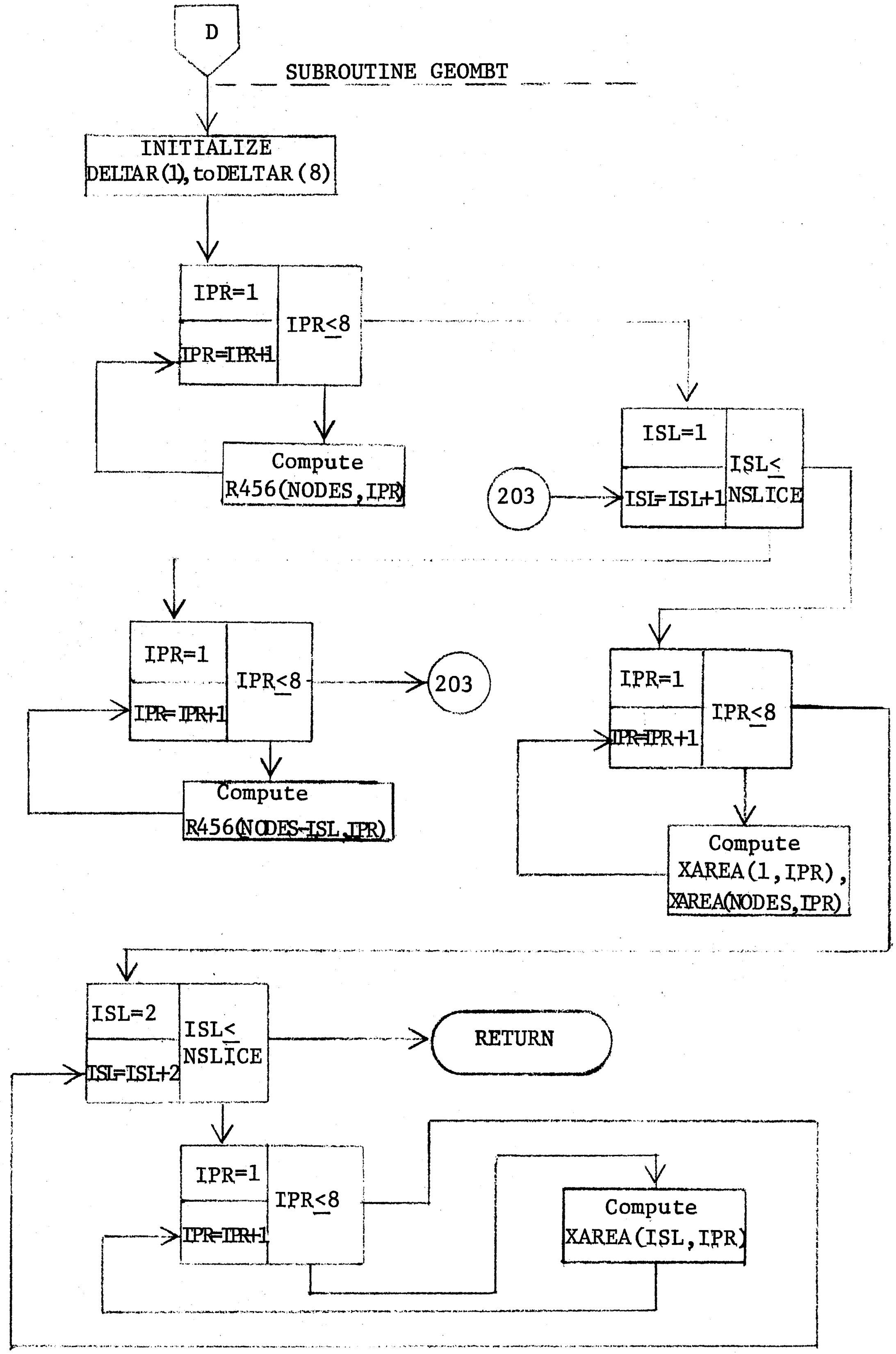
SUBROUTINE GEOM




A
NRUN>199
Yes
INITIALIZE
THICK 1(5) to THICK 8(5),
THICK 1(5) to THICK 8(4),
THICK 1(4) to THICK 8(6),
THICK 1(6) to THICK 8(6),
L(4,1) to L(4,8)
INITIALIZE
THICK 1(5) to THICK 8(5)
THICK 1(4) to THICK 8(4)
THICK 1(6) to THICK 8(4)
L(4,1) to L(4,8)
20
IPR=1
IPR=IPRH
IPR≤8
INITIALIZE
LTC(1),LTC(2)
Compute
L(5,IPR)
L(6,IPR)
NRUN>199
Yes
Compute
LTC(2)
INITIALIZE
LTC(3),LTC(4)
NRUN>199
Yes
Compute
LTC(4)
B
Compute
LTC(5)

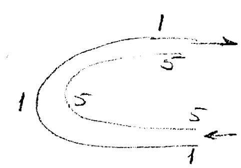

B
NRUN>199
Yes
Compute LTC(5)
Compute LTC(6)
NRUN>199
Yes
LTC(6)
Compute LTC(7)
NRUN>199
Yes
Compute LTC(7)
Compute LTC(8),LTC(9) LTC(10),LTC(11)
NRUN>199
Yes
Compute LTC(11)
C


C
Compute
LEND(ITSECT)
LTOTAL(ITSECT)
RETURN
SUBROUTINE GEOMST
Compute
DELR(1)
I=1
I=I+1
I≤
NSLICE
Compute
R(I+1)
IPR=1
IPR=IPR+1
IPR≤8
Compute
XAREA(1,IPR)
XAREA(NODES,IPR)
I=2
I=I+1
I≤
NSLICE
105
RETURN
D
IPR=1
IPR=IPR+1
IPR≤8
105
Compute
XAREA(I,IPR)


D
SUBROUTINE GEOMBT
INITIALIZE DELTAR(1), to DELTAR(8)
IPR=1
IPR=IPR+1
IPR≤8
Compute R456(NODES,IPR)
ISL=1
ISL=ISL+1
ISL≤ NSLICE
203
IPR=1
IPR=IPR+1
IPR≤8
203
Compute R456(NODES-ISL,IPR)
IPR=1
IPR=IPR+1
IPR≤8
Compute XAREA(1,IPR), XAREA(NODES,IPR)
ISL=2
ISL=ISL+2
ISL≤ NSLICE
RETURN
IPR=1
IPR=IPR+1
IPR≤8
Compute XAREA(ISL,IPR)

```
0001          SUBROUTINE GEOM
0002          COMMON/READ1/TROOM,TCYAVG,PATM,VOLTS,TAMPS,MS,MW,NRUN
0003          COMMON /GEOM1/ XAREA(11,8),R(11  ),L(13,8),LTC(11),LEND(2),DELTAR(
             18),R456(11,8),LIN(2),LOUT(2),LTOTAL(2)
0004          COMMON/GEOM2/RBENDS,RBENDL,DOUT,DIN,PHI,DPHI,DELR,NODES,NSLICE
0005          COMMON /MAIN1/ IST,KOUNT
0006          DIMENSION THICK1(6),THICK2(6),THICK3(6),THICK4(6),THICK5(6)
             1,THICK6(6),THICK7(6),THICK8(6)
0007          REAL * 4 L,LTC,LEND,MW,MS,LIN,LOUT,LTOTAL
0008          ITSECT=1
0009          IF(NRUN.GT.199)  ITSECT =2
0010          NSLICE=10
0011          LIN(1)=12.0+8.0
0012          LIN(2)=20.0+8.0
0013          LOUT(1)=12.0+5.5
0014          LOUT(2)=20.0+5.5
0015          RBENDS=4.75      A
0016          RBENDL=9.875     B
0017          DOUT = 0.875
0018          DIN = 0.771
0019          THICK=0.052
0020          NODES= NSLICE + 1
0021          PHI = 2*3.1416
0022          DPHI = PHI/8.0
0023          PIE14 = 1.0/4.0*3.1416
0024          PIE38 = 3.0/8.0*3.1416
0025          DO 10 IPR=1,8
0026          L(1,IPR )=8.0
0027          L(2,IPR )=6.0
0028          L(3,IPR) =3.0
0029          L(7,IPR) =3.0
0030          L(8,IPR )=6.0
0031          L(9,IPR )=6.0
0032          L(10,IPR)=6.0
0033          L(11,IPR)=8.0
0034       10 CONTINUE
0035          IF (NRUN.GT.199) GO TO 101
0036          THICK1(5) = 0.05665
0037          THICK2(5) = 0.054325
0038          THICK3(5) = 0.052
0039          THICK4(5) = 0.05005
0040          THICK5(5) = 0.0481
0041          THICK6(5) = THICK4(5)
0042          THICK7(5) = THICK3(5)
0043          THICK8(5) = THICK2(5)
0044          THICK1(4) =  (THICK1(5)+THICK)/2.0
0045          THICK2(4) =  (THICK2(5)+THICK)/2.0
0046          THICK3(4) =  (THICK3(5)+THICK)/2.0
0047          THICK4(4) =  (THICK4(5)+THICK)/2.0
0048          THICK5(4) =  (THICK5(5)+THICK)/2.0
0049          THICK6(4) =  (THICK6(5)+THICK)/2.0
0050          THICK7(4) =  (THICK7(5)+THICK)/2.0
0051          THICK8(4) =  (THICK8(5)+THICK)/2.0
0052          THICK1(6) =  (THICK1(4)+THICK)/2.0
0053          THICK2(6) =  (THICK2(4)+THICK)/2.0
0054          THICK3(6) =  (THICK3(4)+THICK)/2.0
0055          THICK4(6) =  (THICK4(4)+THICK)/2.0
0056          THICK5(6) =  (THICK5(4)+THICK)/2.0
```

```
      THICK6(5) =  (THICK6(4)+THICK)/2.0
      THICK7(6) =  (THICK7(4)+THICK)/2.0
      THICK8(6) =  (THICK8(4)+THICK)/2.0
      L(4,1) = (RBENDS-DIN/2.0-THICK1(4)/2.0)*PIE38
      L(4,3)=RBENDS*PIE38
      L(4,5) = (RBENDS+DIN/2.0+THICK5(4)/2.0)*PIE38
      L(4,2)=(L(4,1)+L(4,3))/2.0
      L(4,4)=(L(4,3)+L(4,5))/2.0
      L(4,6)=L(4,4)
      L(4,7)=L(4,3)
      L(4,8)=L(4,2)
      DO 11 IPR=1,8
      L(12,IPR)=41.0
      L(13,IPR)=29.0
   11 CONTINUE
      GO TO 102
  101 CONTINUE
      THICK1(5) = 0.05415
      THICK2(5) = 0.053075
      THICK3(5) = 0.052
      THICK4(5) = 0.051025
      THICK5(5) = 0.05005
      THICK6(5) = THICK4(5)
      THICK7(5) = THICK3(5)
      THICK8(5) = THICK2(5)
      THICK1(4) =  (THICK1(5)+THICK)/2.0
      THICK2(4) =  (THICK2(5)+THICK)/2.0
      THICK3(4) =  (THICK3(5)+THICK)/2.0
      THICK4(4) =  (THICK4(5)+THICK)/2.0
      THICK5(4) =  (THICK5(5)+THICK)/2.0
      THICK6(4) =  (THICK6(5)+THICK)/2.0
      THICK7(4) =  (THICK7(5)+THICK)/2.0
      THICK8(4) =  (THICK8(5)+THICK)/2.0
      THICK1(6) =   THICK1(4)
      THICK2(6) =   THICK2(4)
      THICK3(6) =   THICK3(4)
      THICK4(6) =   THICK4(4)
      THICK5(6) =   THICK5(4)
      THICK6(6) =   THICK6(4)
      THICK7(6) =   THICK7(4)
      THICK8(6) =   THICK8(4)
      L(4,1) = (RBENDL-DIN/2.0-THICK1(5)/2.0)*PIE38
      L(4,3)=RBENDL*PIE38
      L(4,5) = (RBENDL+DIN/2.0+THICK5(4)/2.0)*PIE38
      L(4,2)=(L(4,1)+L(4,3))/2.0
      L(4,4)=(L(4,3)+L(4,5))/2.0
      L(4,6)=L(4,4)
      L(4,7)=L(4,3)
      L(4,8)=L(4,2)
      DO 20 IPR=1,8
      L(12,IPR)=28.0
      L(13,IPR)=16.0
   20 CONTINUE
  102 CONTINUE
      DO 21 IPR=1,8
      L(5,IPR)=L(4,IPR)/PIE38*PIE14
      L(6,IPR)=L(4,IPR)
   21 CONTINUE
```



```
      LTC(1)=4.0
      LTC(2)=LTC(1) +48.0
      IF(NRUN.GT.199) LTC(2 )= LTC(1)+35.0
      LTC(3)=LTC(2) +6.0
      LTC(4)=LTC(3) +3.1416*RBENDS/4.0
      IF(NRUN.GT.199) LTC(4)= LTC(3)+ 3.1416*RBENDL/4.0
      LTC(5)=LTC(4) +3.1416*RBENDS/4.0
      IF(NRUN.GT.199) LTC(5 )= LTC(4)+3.1416*RBENDL/4.0
      LTC(6)=LTC (5)+3.1416*RBENDS/4.0
      IF(NRUN.GT.199) LTC(6 )= LTC(5)+ 3.1416*RBENDL/4.0
      LTC(7)=LTC(6) +3.1416*RBENDS/4.0
      IF(NRUN.GT.199) LTC(7 )= LTC(6)+3.1416*RBENDL/4.0
      LTC(8)=LTC(7) +6.0
      LTC(9)=LTC(8) +6.0
      LTC(10)=LTC(9) +6.0
      LTC(11)=LTC(10)+36.0
      IF (NRUN.GT.199)  LTC(11) = LTC(10) + 23.0
      LEND(ITSECT) = LTC(11)+4.0
      LTOTAL(ITSECT)=LOUT(ITSECT)+LIN(ITSECT)+LEND(ITSECT)
      RETURN
      ENTRY GEOMST
      DELR = (DOUT-DIN)/2.0/NSLICE
      R(1) = DOUT/2.0
      DO 103  I=1,NSLICE
  103 R(I+1)=R(I)-DELR
      DO 104 IPR=1,8
      XAREA (1,IPR)=(R(1)-DELR/4.0)*DPHI*DELR/2.0
  104 XAREA(NODES,IPR)=(R(NODES)+DELR/4.0) *DPHI*DELR/2.0
      DO 105 I=2,NSLICE
      DO 105 IPR=1,8
  105 XAREA (I,IPR)= R(I)*DPHI*DELR
      RETURN
      ENTRY GEOMBT
      DELTAR(1) = THICK1(IST)/NSLICE
      DELTAR(2) = THICK2(IST)/NSLICE
      DELTAR(3) = THICK3(IST)/NSLICE
      DELTAR(4) = THICK4(IST)/NSLICE
      DELTAR(5) = THICK5(IST)/NSLICE
      DELTAR(6) = THICK6(IST)/NSLICE
      DELTAR(7) = THICK7(IST)/NSLICE
      DELTAR(8) = THICK8(IST)/NSLICE
      DO 202 IPR=1,8
  202 R456(NODES,IPR) = DIN/2.0
      DO 203 ISL=1,NSLICE
      DO 203 IPR=1,8
  203 R456(NODES-ISL,IPR)=R456(NODES-ISL+1,IPR)+DELTAR(IPR)
      DO 204 IPR=1,8
      XAREA(1,IPR) = (R456(1,IPR)-DELTAR(IPR)/4.0)*DPHI*DELTAR(IPR)/2.0
  204 XAREA(NODES,IPR)=(R456(NODES,IPR)+DELTAR(IPR)/4.0)*DPHI*DELTAR(IPR
     1)/2.0
      DO 205 ISL=2,NSLICE
      DO 205 IPR=1,8
  205 XAREA(ISL,IPR)=R456(ISL,IPR)*DPHI*DELTAR(IPR)
      RETURN
      END
```

NOMENCLATURE FOR COMPUTER PROGRAM

AID	- cross section of tube, ft^2
AMPS	- fraction of the total current flowing through each segment of each slice of the tube wall, amps
COND	- thermal conductivity of water, Btu/(hr-ft-^oF)
CONDK	- thermal conductivity of Inconel 600, Btu/(hr-ft-^oF)
DELR	- radius increment, inches
DELTAR	- thickness of one slice, inches
DIN	- inside diameter of tube, inches
DOUT	- outside diameter of tube, inches
DPHI	- $\pi/4$
GW	- mass velocity, lbm/(hr-ft^2)
HAV	- average heat transfer coefficient at each station, Btu/(hr-ft^2-^oF
HEF	- Eagle-Ferguson heat transfer coefficient, Btu/(hr-ft^2-^oF)
HSTATE	- Sieder-Tate heat transfer coefficient, Btu/(hr-Ft^2-^oF)
HTCOFF	- peripheral heat transfer coefficient, Btu/(hr-Ft^2-^oF)

HTDB	- Dittus-Boelter heat transfer coefficient, Btu/(hr-ft^2-^{o}F)
IMINS	- peripheral index
IPLUS	- peripheral index
IPR	- peripheral index
ISL	- slice number
IST	- station number
ITSECT	- index for small (1) and large (2) test section
KOUNT	- counter
L	- heating length, inches
LEND	- length of test section, inches
LTC	- distance between each station and start of heating, inches
MW	- mass flow rate, lbm/hr
NMINS	- peripheral index
NPLUS	- peripheral index
NODES	- number of slices
NRUN	- run number
OHMS	- electrical resistance
PHI	- 2π
PIE14	- $\pi/4$
PIE38	- $3\pi/8$
POWER	- power generated in each segment, Btu/hr
PRNO	- Prandtl number

QBALNC	- heat output = mC_p $(T_{out}-T_{in})$ Btu/hr
QFLAXAV	- average heat flux, Btu/(hr-ft^2)
QFLXID	- heat flux at each peripheral position, Btu/(hr-ft^2)
QGEXPT	- heat input = (V)(I), Btu/hr
QLOSS	- heat loss from test section, Btu/hr
QLOST	- heat loss from heat transfer loop, Btu/hr
QPCT	- heat balance error percent
R	- radius to the slice (straight sections), inches
R456	- radius to the slice (bend section), inches
REYNO	- Reynolds number
ROU	- density of water, gm/ml or lbm/ft^3
RSVTY	- electrical resistivity
SPHT	- specific of water, Btu/(lbm-^{o}F)
TAMP	- total current to the test section, amps
TBULK	- fluid bulk temperature, ^{o}F
TCHOK1	- first guess for inside wall temperature, ^{o}F
TCHCK2	- inside surface temperature, ^{o}F
THICK	- wall thickness of tube, inches
TIN	- inlet bulk temperature of water, ^{o}F

TIS	= average inside surface temperature, ^{o}F
TISURF	- inside surface temperature, ^{o}F
TOSURF	- outside surface temperature, ^{o}F
TOUT	- outlet bulk temperature of water, ^{o}F
TROOM	- room temperature, ^{o}F
TWALL	- nodal temperatures of wall, ^{o}F
V	- water velocity, ft/sec
VISC	- water viscosity, lbm/(ft-hr)
VISC	- water viscosity at wall temperature, lbm/(ft-hr)
VOLTS	- voltage drop in test section, volts.

VITA

Mahmood Moshfeghian

Candidate for the Degree of

Master of Science

Thesis: THE EFFECT OF A 180° BEND ON TURBULENT HEAT TRANSFER COEFFICIENT IN A PIPE

Major Field: Chemical Engineering

Biographical:

Personal Data: Born in Shiraz, Iran, June 2, 1951, the son of Ali-Akbahr and Johnbibi Moshfeghian.

Education: Graduated from Sadat High School, Bushehr and Razi High School, Shiraz, Iran, in 1970; received the Bachelor of Science degree in Chemical Engineering from the Oklahoma State University in May, 1974; completed the requirements for the Master of Science degree in Chemical Engineering in July, 1975.

Professional Experience: Graduate Teaching Assistant, School of Chemical Engineering, Oklahoma State University, 1974-75.